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SUMMARY

The effect of adding steam on thermodynamic performance of two Brayton gas turbine cycles was analytically investigated. The first cycle was a simple one in which steam was generated in a heat recovery boiler and then injected into the combustor. The other was a recuperated cycle with an evaporator at the compressor exit. The addition of steam resulted in increased cycle efficiency and specific power. Results are presented for a range of turbine inlet temperatures and compressor pressure ratios. Component efficiencies and pressure losses typical of those in a stationary powerplant were used.

The simple cycle efficiency increased with the amount of steam generated until a maximum cycle efficiency was achieved. Further increases in steam resulted in decreased efficiency. Adding steam resulted in the maximum cycle efficiency increasing between 25 and 30 percent. Specific power increased continuously with increased steam addition. At the steam addition rate for maximum efficiency the increase in specific power in terms of equivalent-air flow rate at the turbine inlet was between 25 and 35 percent, and the increase in terms of compressor inlet air flow was between 50 and 90 percent for the same steam addition rates.

For the recuperated cycle, the increase in maximum cycle efficiency with steam addition was less than 3 percent for the cases examined. However, for most cases examined, the specific power at maximum efficiency increased approximately 20 percent when calculated in terms of equivalent-air flow at the turbine inlet and approximately 40 percent when calculated in terms of compressor inlet air flow.

For the assumptions used, the simple cycle with steam addition had a slightly higher maximum efficiency than the recuperated cycle with steam addition. However, at maximum cycle efficiency the specific power was significantly greater for the simple cycle. At the same turbine inlet temperature the heat recovery boiler had a smaller heat-transfer area than the recuperator for the same power output.

The simple cycle had a greater steam addition rate for maximum efficiency than the recuperated cycle. Even for the simple cycle, the water use rates at maximum efficiency were less than the water consumption for a conventional steam powerplant employing wet cooling towers.

INTRODUCTION

This report presents analytic predictions of the performance of gas turbines using steam injection. Steam is generated and injected into the turbine flow so that both the specific power and efficiency are increased. Two cycles were examined. The first is a modified simple gas turbine cycle, and the second is a modified recuperated gas turbine cycle. The modified simple cycle is shown in figure 1. This cycle incorporates a heat recovery boiler to generate steam using the energy available in the turbine exhaust. This is similar to a conventional gas-turbine - steam-turbine combined cycle. But, unlike a combined cycle, steam is injected into the combustor where it is heated along with the compressor discharge air to the desired turbine inlet temperature. Because of the added mass flow through the turbine, the specific power is increased over that of a simple cycle without steam addition. Since the steam is generated using exhaust heat which would otherwise be wasted, the efficiency is also increased.

The concept of using steam or water to augment the turbine flow is not new; discussions of this concept can be found in textbooks such as reference 1. The possibility of improving cycle efficiency has recently led to renewed interest in this cycle. References 2 to 6 discuss the performance of a simple gas turbine cycle to which a steam boiler was added in order to recover the heat available in the turbine exhaust. Although the analytic data available in these references are for just a few selected operating conditions, they indicate that significant gains in efficiency and specific power can be made by using a heat recovery boiler. Reference 6 discusses the benefits of reduced NO_x emissions resulting from steam addition as well as operational considerations for gas turbines with steam injection. Reference 7 discusses the satisfactory operating experiences when process steam was used to increase the specific power for a gas turbine.

The second cycle analyzed was a recuperated gas turbine cycle in which an evaporator was added at the outlet of the compressor. This cycle is shown in figure 2. The evaporator generates steam by adding water to the compressor discharge air. The energy to evaporate the water is supplied by the sensible heat of the compressed air. Consequently, the air temperature at the exit of the evaporator is significantly less than it is at the inlet. Since the air temperature at the recuperator inlet is decreased, the exhaust temperature at the recuperator outlet can be reduced.

This results in increased recovery of the energy from the exhaust gases and, consequently, increased cycle efficiency. Because of the increased mass flow through the turbine, the specific power is also increased. This cycle is described in reference 1. This cycle has also received renewed interest because of possible performance gains. References 8 to 10 discuss the cycle and show that significant performance improvements can be achieved by adding an evaporator. The analytic data in these references are for a few selected operating conditions.

The purposes of this report are to examine the effects of steam addition on the performance of simple and recuperated cycles and to make consistent comparisons of the performance of these two cycles with gas turbine cycles that do not use steam addition. Analytic results are presented for a wide range of turbine inlet temperatures, pressure ratios, and steam to air ratios. The range of turbine inlet temperatures examined was from 816° to 1093° C (1500° to 2500° F). For each turbine inlet temperature appropriate ranges of pressure ratios and steam to air ratios were examined. The results incorporate assumptions for the losses which would typically be present in a stationary powerplant. The efficiency of the turbomachinery was accounted for, and estimates were used for mechanical losses. Pressure drop losses associated with the different components of the cycle were included in the analysis. The sensitivity of cycle performance to variations in the assumed values for these losses was examined. Data are presented for both uncooled and air-cooled turbines. Several cases are examined in detail to show state points within the cycle and to compare state-point changes associated with steam addition.

In addition to cycle performance, two other topics are discussed. The heat-transfer area required for the simple cycle heat recovery boiler is compared with the required recuperator heat-transfer area. The thermodynamic considerations associated with the recovery of the water used to generate the steam are also discussed.

CYCLE ANALYSIS AND ASSUMPTIONS

A description of the analysis to determine the performance of both simple and recuperated cycles with steam addition is given herein. Also, the method used to calculate the required amount of turbine cooling is discussed.

Simple Cycle with Steam Addition

Figure 1 shows a schematic diagram of a simple open Brayton thermodynamic cycle with steam addition. Turbine exhaust heat is used to generate steam in a heat recovery boiler. The steam is injected into the burner along with a portion of the compressor discharge air. Fuel is burned to achieve the desired turbine inlet temperature at the burner exit. The amount of compressor discharge air which does not go to the burner is determined by turbine cooling requirements.

The performance results depend on several assumptions. The following is a brief description of the calculation procedure and assumptions used in the analysis. The work required to compress the inlet air is calculated using the specified pressure ratio, an assumed value for the compressor polytropic efficiency, and the specific heat ratio at the average of the compressor inlet and outlet temperatures. The values for the compressor polytropic efficiency and other parameters used in the analysis are given in table I. Some parameters were investigated over a range of values while others were generally held constant. The effect on cycle performance resulting from variations in these parameters will be discussed. Because a constant polytropic efficiency was assumed, the adiabatic efficiency varied with pressure ratio. Part of the compressor discharge air is used for turbine cooling. Although complete combustion in the burner is assumed, a combustion heat loss of 1 percent (see table I) is used in the analysis. The actual composition of the gases flowing through the compressor, turbine, and heat recovery boiler are used to calculate the cycle performance. The fuel is assumed to be a pure hydrocarbon with a hydrogen to carbon ratio typical of kerosene. The calculations use the lower heating value for the fuel. If there is no turbine cooling, the work done by the turbine is found from the turbine pressure ratio, a turbine polytropic efficiency, and the specific heat ratio at the average of the turbine inlet and outlet temperatures. When turbine cooling is present, the work is found from the enthalpy drop across each turbine stage. Since coolant is added to the mainstream flow at the exit of each turbine row, the mass flow rate increases for sucessive stages.

In the thermodynamic analysis of the simple cycle with steam addition, the heat recovery boiler is described by a single parameter. This parameter is the minimum of the approach temperature difference and the pinch-point temperature difference. This minimum ΔT was typically taken as 28° C (50° F) as can be seen in table I.

The minimum ΔT can occur at either of two locations within the heat recovery boiler. Figure 3 shows sketches of the temperature profiles within the boiler assuming a counterflow configuration. Figure 3(a) is a sketch of the temperature profiles for relatively low steam flows. For low steam flows little heat is extracted from the exhaust gas. Consequently, the slope of the exhaust gas temperature line is low, and the resulting steam exiting temperature is the exhaust gas inlet temperature minus the approach temperature difference. The minimum ΔT occurs at the steam exit of the boiler (location 1 in the sketch). For low steam flows, then, the minimum ΔT is equal to the approach temperature difference.

Figure 3(b) is a sketch of the temperature profiles for a somewhat larger steam flow. This sketch has been drawn to show the condition when the approach temperature difference equals the pinch-point temperature difference. For this condition, the minimum ΔT occurs simultaneously at locations 1 and 2 (the start of steam evaporation). Since the increased steam flow requires an increased heat-transfer area, there is an increase in boiler size in going from the condition shown in figure 3(a) to the condition shown in figure 3(b).

Figure 3(c) is a sketch of the temperature profiles when the steam flow has been increased even further. In this condition, the approach temperature difference is greater than the pinch-point temperature difference. For this condition, the minimum ΔT occurs at location 2. Once there is a shift in the location of the minimum ΔT, the temperature at which the steam exits the boiler decreases with increased steam flow for a fixed exhaust gas inlet temperature. The analysis was done by specifying a minimum ΔT and then determining whether it occurred at location 1 (steam exit of the boiler) or location 2 (start of steam evaporation). The steam exit temperature was then calculated by assuming a linear exhaust gas temperature profile and using actual steam properties.

Pressure losses, shown in table I, were assumed for the burner and for both the exhaust and water sides of the boiler. These losses were chosen sufficiently large to account for the pressure drops in the lines between the components. No separate pressure losses were included for the lines between the components. A mechanical efficiency, applied to the turbine output, and a generator efficiency, applied to the net output, were used in the analysis. The pressure drops and efficiency values are typical of those used in references 11 and 12.

Recuperated Cycle with Steam Addition

The recuperated cycle with steam addition is shown in figure 2. In this cycle the compressor discharge flow gives up heat in order to generate steam by vaporizing water in an evaporator. As a consequence, this air-water vapor mixture is substantially lower in temperature than the compressor discharge air. For higher compressor pressure ratios, with their corresponding higher compressor discharge temperatures, significant quantities of steam can be generated. With steam addition, the air-water vapor mixture entering the recuperator is at a lower temperature than without steam addition. Therefore, more heat can be recovered from the exhaust in the recuperator. This results in an increased efficiency. Also, the increased mass flow through the turbine results in increased specific power.

The same assumptions were used for both the simple and recuperated cycles. For the recuperated cycle additional assumptions were made for the properties of the evaporator and recuperator. The assumed values used are also given in table I. The performance calculations for both the simple and recuperated cycles were done by coding the necessary equations in a digital computer program.

Turbine Cooling Requirements

The analysis of the turbine cooling requirements is the same for both the simple and recuperated cycles. The analysis for turbine cooling is based on using compressor discharge air as the turbine coolant. The following procedure is used. The number of turbine stages is selected based on the enthalpy drop across the turbine and a mean blade speed of 366 meters per second (1200 ft/sec). The analysis assumes equal pressure ratios across each stage of the turbine. For each stage, the amount of cooling required for the stator vanes and the rotor blades is calculated. This calculation is based on the correlation given in reference 13 for impingement-convection cooling methods. The following equation was used to determine the amount of cooling needed for the stator and rotor row of each stage:

$$\frac{T_i - T_m}{T_i - T_c} = \frac{1}{1 + 0.13\left(\frac{w_c}{w_i}\right)^{-0.7}} \qquad (1)$$

In this equation, w_c/w_i is the ratio of cooling flow required to the flow entering the stator or rotor row of the stage; T_i the total gas temperature at the inlet of the stator row, or the relative total gas temperature at the inlet of the rotor row; T_m the maximum metal temperature allowed; and T_c the coolant temperature.

All coolant was assumed to be extracted from the compressor exit. Therefore, T_c is the compressor exit temperature. At the exit of each stator and rotor row the gas temperature is adjusted to account for adding the stator or rotor coolant. The adjusted temperature is the equilibrium temperature resulting from perfect mixing of the mainstream flow and the coolant. The coolant flow for the next rotor or stator row is calculated using this adjusted temperature as the inlet gas temperature in equation (1). This process is repeated for all the turbine stages. In addition to the coolant flow given by equation (1), an additional blockage flow is used for cooled turbines. This flow was chosen at 0.5 percent of the turbine inlet flow per turbine stage. Such flow is needed to prevent hot combustion products from flowing into the spaces between the blade rows. This blockage flow is assumed not to be mixed with the mainstream flow within the turbine but added at the exit of the last stage.

Specific Power Calculation Procedure

Specific power for gas turbines is generally expressed as power per unit of compressor inlet mass flow rate. However, with steam addition, the flow rate at the turbine inlet can be significantly greater than the flow rate at the compressor inlet. When the flow at the turbine inlet exceeds the compressor flow, expressing specific power in terms of equivalent air at the turbine inlet is a more conservative way of showing the gains in specific power due to steam addition. Throughout this report the specific power is generally given in terms of equivalent air at the turbine inlet. However, comparisons are given for the two ways of expressing the specific power. The equivalent-air flow rate at the turbine inlet is the mass flow rate of air which has the same volumetric flow as the actual turbine inlet flow. This equivalent-air flow uses the actual composition at the turbine inlet. It accounts for both the added steam and fuel flow as well as the density of the gas mixture.

RESULTS

The cycle performance for a range of turbine inlet temperatures (816° to 1093° C (1500° to 2500° F)), pressure ratios, and steam addition rates will be given. Both the cycle efficiency and specific power will be discussed. In addition, detailed state-point data will be given for several cases. These cases represent possible operating points at each turbine inlet temperature. The effect on cycle performance resulting from varying the assumed values will be discussed for both the simple and recuperated cycles. Also, the relative performance between cycles will be discussed. This discussion includes consideration of the required heat exchanger area per unit of power output. In addition to cycle performance, the thermodynamic considerations associated with the recovery of the water used to generate steam will be discussed.

Simple Cycle Performance with Steam Addition

Figure 4 shows the effect on cycle efficiency due to steam addition for a typical case. Without a heat recovery boiler or steam addition, the efficiency at this turbine inlet temperature and pressure ratio is 33.3 percent. The addition of a heat recovery boiler results in increased system pressure drops. Only when the mass ratio of steam to air exceeds 0.01 is there an increase in cycle efficiency. With steam addition the turbine flow is increased, resulting in greater power output. However, the fuel flow also increases. The increased fuel flow is required to heat the steam from the temperature at the exit of the heat recovery boiler to the turbine inlet temperature. The increase in power output due to steam addition is a result of both the added turbine flow and the energy released from the added fuel. The net result is an increase in cycle efficiency. The efficiency increases with increased steam addition until there is a shift in location in the minimum ΔT for the boiler. This shift is from the steam boiler exit to the pinch point within the boiler. After the minimum ΔT shifts, there is a decrease in cycle efficiency with increased steam addition. The turbine exit temperature does not vary greatly with the amount of steam addition. Therefore, prior to the shift in the location of the minimum ΔT, all of the steam enters the burner at approximately the same temperature. However, after the shift, the temperature of the steam decreases with increasing steam addition. As the rate of steam increases and the steam temperature into the burner decreases, more fuel

is required per unit mass of steam just to heat the steam to the same turbine inlet temperature. The net gain in generator output due to increased turbine flow no longer outweighs the increased fuel and there is a decrease in efficiency. Although it is not shown in the figure, the specific power continuously increases with increased steam addition. In some applications it may be desirable to use steam flows greater than those corresponding to maximum efficiency in order to achieve the higher specific power.

The maximum amount of steam which can be generated is limited due to several different causes. Figure 4 shows the steam to air ratios at which the maximum amount of steam addition may be limited due to operational considerations. The steam to air ratio at which there is no excess air is shown in the figure. At this ratio of 0.68 there is no excess air to support additional fuel combustion. Further additions of steam are not possible if the turbine inlet temperature is not reduced. From the standpoint of corrosion avoidance, it is desirable that corrosive constituents in the exhaust gases do not condense in the boiler. Therefore, a positive temperature margin is generally provided to insure that no condensation occurs. In addition to the injected steam, the exhaust gases contain additional steam formed by the combustion process. As a result, the partial pressure of the water vapor is relatively high. Figure 4 shows the steam to air mass ratio (0.35) at which steam will start to condense in the exhaust side of the boiler. If the fuel contains sulfur, sulfuric acid formed in the exhaust gas stream would probably condense before this value is reached. It may also be desirable to have only dry steam enter the burner. The steam to air ratio at which wet steam is injected (0.27) is also shown in figure 4, and this may also represent an operational limit. For steam to air ratios greater than this value, the enthalpy of the steam exiting the boiler decreases with increasing steam addition, even though the temperature of the steam remains constant. The steam to air ratios at which operational limits occur will vary for different turbine inlet temperatures and compressor pressure ratios.

Figure 5 shows the cycle efficiency and specific power as a function of steam addition rates and compressor pressure ratios for three different turbine inlet temperatures. Each of the five parts of figure 5 gives performance results for a specific turbine inlet temperature and cooling assumption. Because of losses, the pressure ratio across the compressor is greater than the pressure ratio across the turbine. The steam addition rate is the mass flow rate ratio of steam to compressor

inlet air. Data are shown in figure 5 for three different turbine inlet temperatures. To show the influence of turbine cooling, data are shown for both an air-cooled turbine and an uncooled turbine. In each part of figure 5 results are shown for a recuperated cycle without steam addition. This is done for reference purposes only. The recuperator data were calculated using an effectiveness of 0.85, while the minimum ΔT for the heat recovery boiler was 28^{o} C (50^{o} F). In figures 5(a) and (b) the recuperated cycle data are given as segmented curves. This was done because the recuperated cycle results were strongly dependent on the number of turbine stages used in the analysis. The number of stages varied with the compressor pressure ratio. It will be shown later that the results for the simple cycle were less dependent on the number of turbine stages used. The data in figure 5(a) are for an uncooled turbine. At this turbine inlet temperature of 816^{o} C (1500^{o} F), the performance of an air-cooled turbine would be nearly identical.

The results for the different turbine inlet temperatures show that, even when specific power is expressed in terms of equivalent air at the turbine inlet, there are significant gains in specific power due to steam addition. Data are shown for a range of steam to air ratios in excess of those needed to produce peak cycle efficiency.

Only a limited range of steam addition rates are shown in figure 5(c). This figure contains data for an air-cooled turbine at the highest inlet temperature considered. At 1371^{o} C (2500^{o} F) the maximum steam to air ratio was limited by the amount of air available in the burner. No further steam could be added without reducing the turbine inlet temperature. The steam addition rate at which this limit was reached decreased with increased pressure ratio because as the pressure ratio increased the coolant temperature increased. Consequently, the amount of turbine coolant air required increased, and the amount of air available for combustion decreased.

The results in figure 5 show that for a given turbine inlet temperature and cooling assumption the maximum efficiency is achieved at a specific steam addition rate and pressure ratio. In a simple cycle without steam addition, one way to improve efficiency is to increase the turbine inlet temperature. However, to achieve the maximum efficiency gain the pressure ratio must also be increased. It is interesting to note that with steam addition the pressure ratio at which maximum efficiency is achieved is less than the pressure ratio at which the simple cycle achieves maximum

efficiency at the same turbine inlet temperature. The maximum cycle efficiency increased between 25 and 30 percent as a result of steam addition for each of the five cases examined. This means that significantly less fuel would be required with steam addition for the same power output at the same turbine inlet temperature. The maximum efficiency for the simple cycle with steam addition was generally greater than the maximum efficiency for the reference recuperated cycle without steam addition. The exception in the cases studied is the 816° C (1500° F) turbine inlet temperature, shown in figure 5(a), where the maximum simple cycle efficiency was about 0.5 percentage point less than the maximum efficiency of the reference recuperated cycle. For the other turbine inlet temperatures, the maximum cycle efficiency was up to 4 percentage points greater than the maximum efficiency of the recuperated cycle at the same turbine inlet temperature.

The specific power increases with increasing steam addition, even for steam addition rates greater than that for maximum efficiency. Figure 5 shows that specific power at maximum efficiency for the simple cycle with steam addition was between 25 and 35 percent greater than the specific power at the same pressure ratio but without steam addition. When the specific power was calculated in terms of compressor inlet air, the corresponding increase in specific power due to steam addition was between 50 and 90 percent. The lowest percent increase was associated with the lowest steam to air ratio, and the highest percent increase was associated with the highest steam to air ratio. While the maximum efficiency for the simple cycle with steam addition is not much greater than the maximum efficiency for the recuperated cycle shown, the specific power for the simple cycle with steam addition is significantly greater than that for the recuperated cycle.

It is of interest to compare the performance of the simple cycle with steam addition to that of a gas-turbine - steam-turbine combined cycle. Both cycles employ a heat recovery boiler. But, in the combined cycle the steam generated is used in a conventional steam turbine which produces additional output power. The simple cycle does not use a separate turbine, nor is a condenser needed. Reference 14 describes combined cycles in further detail, and it gives some of their performance characteristics. Comparisons between the simple and combined cycles are only approximate, since the perfomance for the two cycles was not calculated with the same assumptions. The data in reference 14 show that at the same turbine inlet temperature the combined cycle had a cycle efficiency several percentage points greater.

When the two cycles were compared at the same turbine inlet temperature, they both had about the same specific power.

Figure 6 shows the temperature of the exhaust gas at the exit of the heat recovery boiler as a function of the steam to air mass flow rate ratio. Each part of figure 6 has data for the same turbine inlet temperature and cooling assumption as the corresponding part of figure 5. The results are similar for each of the five parts of figure 6. For low steam to air ratios the slope of the exhaust gas exit temperature curve is very steep. However, for high ratios, where the minimum ΔT occurs at the pinch point, the slope is much less. In each part of figure 6 the curves for the different pressures ratios do not exactly converge at high steam to air ratios. However, the temperature difference between the curves is very small, and only one line is shown for clarity. If the exhaust gas temperature is sufficiently low, and the amount of water vapor in the exhaust is relatively high, water vapor will condense near the exit of the boiler. The temperature at which this occurs is also shown in figure 6. Because this occurs for high steam to air ratios, and at a location in which the exhaust stream has already passed the pinch point, cycle performance is not affected by water vapor condensation near the exit of the exhaust stream from the heat recovery boiler. From each part of figure 6 it can be seen that a lower pressure ratio results in a higher exhaust gas exit temperature at the same steam to air ratio.

Cycle parameters. - Table II presents the temperatures, pressures, and relative mass flow rates at different points in the cycle for six selected cases. These simple cycle cases are labeled S1 to S6. The first five cases correspond to the turbine inlet temperature and cooling assumptions used in the five parts of figure 5. These cases were chosen as possible operating points at different turbine inlet temperatures. The sixth case is included for comparison and is for the same turbine inlet temperature and pressure ratio as the fourth case, but with no steam addition. The five cases with steam addition shown in table II were chosen from the data in figure 5 with a pressure ratio slightly less than that for maximum efficiency. At this lower pressure ratio the specific power was slightly greater than the specific power corresponding to maximum efficiency. The steam addition rate was then chosen to give the maximum efficiency for the chosen pressure ratio.

It can be seen from table II that the boiler exhaust outlet temperatures are quite low for the five cases with steam addition compared with the turbine exhaust tem-

perature for the case without steam addition. This shows that with steam addition a significant quantity of exhaust gas energy can be recovered in the heat recovery boiler. In table II the boiler exhaust outlet temperature is shown to be lower for the higher turbine inlet temperature cases. This is a result of the higher steam addition rates that can be attained with the higher turbine inlet temperatures before the pinch-point limit is reached in the heat recovery boiler. At the highest turbine inlet temperature the exhaust outlet temperature is only about 28° C (50° F) greater than the dew point temperature for the water vapor in the exhaust. Temperatures this close to the dew point could result in corrosion problems because of the acid formation caused by fuel impurities.

Two measures of specific power are shown for each case in table II. For the case of no steam addition, case S6, there is little difference between the specific power whether expressed in terms of compressor inlet air flow or in terms of equivalent air at the turbine inlet. For case S3, with the high turbine inlet temperature of 1371° C (2500° F), the specific power is only about 10 percent less when expressed in terms of equivalent air at the turbine inlet. The mass flow at the turbine inlet is equal to the compressor inlet flow plus the steam and fuel flows less the turbine coolant flow. For this case the added flow of the steam and fuel is largely offset by the turbine coolant flow. For the other cases, however, there are larger differences in the two expressions for specific power. It can be seen that the gains in specific power due to steam addition in terms of compressor inlet air are generally significantly higher than the gains in terms of equivalent air at the turbine inlet.

The water use rates for the cases shown in table II varied between 1.06 and 1.67 kilograms per kilowatt-hour (0.28 and 0.44 gal/kW-hr). These values are only about half the water consumption rate for a conventional coal-fired steam plant employing wet cooling towers (ref. 6).

Effect of turbine inlet temperature. - Figure 7 gives cycle efficiency as a function of turbine inlet temperature for constant steam addition rates and pressure ratios. The steam addition rates are the same as those given for the cases shown in table II. Figure 7(a) assumes an uncooled turbine, and figure 7(b) is for an air-cooled turbine. For both cooling assumptions the efficiency continuously increases with increasing pressure ratio at higher turbine inlet temperatures. At lower tur-

bine inlet temperatures the curves for the higher pressure ratios cross those for a lower pressure ratio at the same steam to air ratio.

From figure 7(a) it can be seen that without steam addition the efficiency increases continuously with increased turbine inlet temperature for the uncooled turbine assumption. From figure 7(b) it can be seen that there is a relatively flat peak in efficiency at constant pressure ratio for an air-cooled turbine without steam addition. This is a result of the increased cooling required as the turbine inlet temperature is increased. The results with steam addition show a markedly different behavior. From figures 7(a) and (b), it can be seen that at a fixed steam to air ratio, the efficiency peaks at a specific turbine inlet temperature. This peak is sharply defined for the uncooled turbine assumption as is seen in figure 7(a). When the turbine inlet temperature is to the left of the peak, the minimum ΔT of the heat recovery boiler is located at the pinch point (start of boiling). Under this circumstance, the approach temperature difference is greater than the minimum ΔT. As the turbine inlet temperature increases, the approach temperature difference decreases until it equals the pinch-point temperature difference. The approach temperature difference equals the pinch-point temperature difference at peak efficiency. As the turbine inlet temperature increases, the turbine exit temperature increases. To the left of the peak, the energy of the steam exiting the boiler increases because of both the higher turbine exhaust temperature and the reduced approach temperature difference as the turbine inlet temperature is increased. For turbine inlet temperatures to the right of the peak, the minimum ΔT is equal to the approach temperature difference. In this region, increasing the turbine inlet temperature causes the steam exiting the boiler to increase in temperature only as much as the turbine exit temperature is increased.

The data for the air-cooled turbine in figure 7(b) do not show the same discontinuity in slope. As seen in figure 7(a), for the uncooled turbine assumption, the rate of increase in efficiency for turbine inlet temperatures to the left of the peak is almost constant. But for an air-cooled turbine, the required cooling increases with higher inlet temperatures. This causes the slope of the efficiency curve to decrease with increased turbine inlet temperatures. The result is a smooth curve of efficiency as a function of turbine inlet temperature for an air-cooled turbine.

Perturbation analysis. - The results shown in figure 5 depend on assumed values for several independent cycle parameters. Among these parameters are compo-

nent pressure drops, compressor and turbine efficiencies, and the heat recovery boiler minimum temperature difference. An analysis was done to determine the effect of variations in several parameters on the performance for the cases given in table II. The effect of small changes in the assumptions on the cycle efficiency and specific power can be seen from the information presented in tables III and IV. Influence coefficients were determined for independent variables using the following procedure. First, a normalized independent variable X was defined for each independent parameter x. Table III describes each independent parameter, gives their value for each of the six base cases, and shows the definition of the normalized independent variables. Next, numerical approximations were calculated for the partial derivitives of efficiency and specific power for each of the independent variables at each of the six base cases. The approximations to the partial derivatives were determined by calculating each base case, but with each independent parameter changed in turn by a few percent. In addition, checks were made to insure that the perturbations were by the proper amounts. Table IV gives the values of these partial derivatives for each case. The values of the partial derivatives are the influence coefficients for each of the independent variables on efficiency and specific power. In table IV the dependent variables are denoted by Y. The influence coefficient for cycle efficiency is the percentage point (not percent of base value) change in efficiency per percent change in the independent variable. The influence coefficient for specific power is, however, the percent change in specific power per percent change in the independent variable. The specific power given in these tables is expressed in terms of equivalent air at the turbine inlet.

It has already been shown that the effect of steam flow rate ratio, turbine inlet temperature, and compressor pressure ratio on cycle efficiency changes abruptly with a shift in location of the minimum boiler ΔT. Table IV gives the influence coefficient on each side of the location of the shift for variables which behave in this manner. The normalized independent variables are defined so as to be zero at the discontinuity. The other independent variables had slight variations in their influence coefficients from one side of the base case to the other. The average value of the influence coefficients is given for these variables so that the numerical approximation for the influence coefficient is centered at the base case.

To illustrate the use of the data in table IV, it can be seen that for case 1 the efficiency influence coefficient is 0.49 for the compressor polytropic efficiency.

This means that an increase of 1 percentage point in the compressor polytropic efficiency would result in the cycle efficiency increasing by 0.49 percentage point. Also, the influence coefficient for the effect of compressor efficiency on specific power is 2.24. Thus, a 1 percentage point increase in compressor efficiency would result in the specific power increasing by 2.24 percent. The significance of the magnitude of the influence coefficients given in table IV is related to the definitions of the normalized independent variables given in table III. For example, in case 1, the influence coefficient for the minimum ΔT is given as -0.01. This seems to be a small value. However, from the definition of the normalized variable in table III, it is seen that doubling the minimum ΔT results in a 100-percent change in x. The corresponding change in X would be 100. Therefore, doubling the minimum ΔT from 28° to 56° C (50° to 100° F) would result in the cycle efficiency decreasing by 1.0 percentage point. At the same time the specific power would increase by 0.1 percent.

Most influence coefficients do not show large variations among the five base cases which have steam addition. The influence coefficients tend to increase with decreasing cycle efficiency. Steam injection has an effect on the relative influence of the other components within the cycle. This can be seen by comparing the data in table IV for case S4 with steam addition and case S6 with no steam addition. The influence on cycle performance of parameters which affect compressor work are seen to be less for the case with steam addition. This is because, with steam addition, the compressor work is a smaller fraction of the turbine work. For case S4 the influence coefficient for compressor polytropic efficiency on cycle efficiency is 0.36. For specific power the influence coefficient is 1.52. The corresponding case with no steam addition, case S6, has influence coefficients of 0.55 and 2.55. It should be noted that cases S4 and S6 were calculated at the same pressure ratio (16). However, at the same turbine inlet temperature as case S6 the cycle without steam addition achieves maximum efficiency at a higher pressure ratio. Although not shown in table IV, the influence coefficients for case S6, with the pressure ratio changed to 20, are 0.67 and 3.45. This shows that at near maximum efficiency the influence of compressor polytropic efficiency on cycle performance is only about half as great for the cycle with steam addition. There are data showing the effects of turbine cooling for cases S4, S5, and S6 in table IV, even though these cases assume an uncooled turbine. The reason for this is that the parameter "Any cooling" shows the

influence of changing the turbine cooling assumption. For cases S2 and S3 the results show the effect of assuming an uncooled turbine. For the other cases this parameter shows the effect of including turbine cooling. The influence coefficients for different cases at the same turbine inlet temperature do not have the same magnitude. The magnitudes are different because of different pressure ratios and steam to air mass flow rate ratios. The other influence coefficients associated with turbine cooling are always given for an air-cooled turbine.

Some caution should be used in applying the influence coefficients over a wide range of parameter variation. The data in tables II and IV can be used to indicate the errors incurred in making predictions over wide ranges. If influence coefficients are taken at one case, predictions can be made for the other cases. Predictions were made using the influence coefficients which took into account changes in turbine inlet temperature, pressure ratio, and steam flow rate ratio. When going from one case in table II to another with a turbine inlet change of 278° C (500° F), the predicted values of specific power were within 10 percent of the actual values of specific power. The predicted values for cycle efficiency were not in good agreement. However, this was expected since the data in table II are for optimum steam flow rate ratios and near-optimum pressure ratios. The influence coefficients should be used to indicate trends in the data and not exact values at new cycle conditions.

Recuperated Cycle Performance with Steam Addition

Figure 8 gives the cycle efficiency and evaporator exit temperature as a function of the amount of water evaporated for a recuperated cycle. The high pressure ratio (12) was chosen to maximize the benefits obtained from using an evaporator. The data show increasing cycle efficiency as a function of the amount of water evaporated. Also shown is the temperature of the air-water vapor mixture leaving the evaporator. Because the heat used to evaporate the water comes from the compressor discharge air, the exit temperature decreases uniformly as the amount of evaporated water increases. As can be seen from the figure, this decrease in temperature is substantial. An additional abcissa scale is shown. This scale gives the relative humidity of the air leaving the evaporator. Because of the temperature decrease with increased evaporation, the relative humidity varies nonlinearly with the amount

of water evaporated. When the relative humidity reaches 100 percent, no further water can be added to the air stream by evaporation. Although not shown on the figure, the specific power increases in proportion to the amount of water evaporated.

Figure 9 gives the efficiency and specific power for recuperated cycles with and without an evaporator to generate steam. Figures 9(a) to (e) give results for the same turbine inlet temperature and cooling assumption as was used for the simple cycle with steam addition data shown in figure 5. The data in figure 9 were calculated using a recuperator effectiveness of 85 percent. The cycle performance is given for a range of pressure ratios. The results shown for the steam addition cases were calculated assuming that the air from the evaporator was completely saturated. In each part of figure 9 it can be seen that at the lower pressure ratios the efficiency of the cycle without steam addition is greater than the efficiency of the cycle with the evaporator present. At low pressure ratios little water is evaporated because of the relatively low compressor exit temperatures, but there is a pressure drop loss caused by the presence of the evaporator. In these cases the beneficial effect of the steam addition to the turbine flow is outweighed by the increased pressure losses. With steam addition, the maximum cycle efficiency was increased. For the data shown, the maximum efficiency with steam addition was about 1 percentage point higher than that obtained without steam addition. This would result in only a few percent fuel savings. The maximum efficiency for the cycle with steam addition occurs at a pressure ratio significantly greater than the pressure ratio corresponding to the maximum efficiency of the cycle without steam addition. Also, the cycle efficiency varied less with variation in pressure ratio for the cycles with steam addition than the corresponding cycles without steam addition. The data in figures 9(b) and (c) show the effect of the number of turbine stages on cycle efficiency. The number of turbine stages strongly influences the amount of turbine coolant required. At 1371° C (2500° F), shown in figure 9(c), the number of turbine stages has the greatest effect on cycle performance for the cases examined. When the number of stages is increased, there is additional coolant flow. This increased flow is needed for the blockage flow for the additional stage, and increased vane and blade coolant is needed because of the different gas temperature profile within the turbine. Since the coolant flow bypasses the evaporator, the increased coolant results in a lower amount of steam generated. This also contributes to a

lower cycle efficiency. At 1093° C (2000° F) the amount of turbine coolant is less. Consequently, the effect of the number of turbine stages is less. The data in each part of figure 9 show the increase in efficiency at the pressure ratios corresponding to the maximum efficiency for the cycle with steam addition to be approximately 2 percentage points. Also, the dropoff in efficiency with increased pressure ratio is less for the cycle with steam addition. This would be significant in a retrofit application where the recuperated cycle without steam addition had a pressure ratio greater than that for maximum efficiency.

Even though the gains in cycle efficiency due to steam addition were smaller than for the simple cycle data shown in figure 5, there were significant increases in specific power. Except for the air-cooled turbine at 1371° C (2500° F) case, the increases in specific power at maximum efficiency for the cases presented were approximately 20 percent. For the air-cooled turbine at 1371° C (2500° F) the pressure ratio at maximum effficiency was approximately the same with and without steam addition. The specific power increased only about 10 percent when the specific power at maximum efficiency for steam addition was compared to the specific power at the same pressure ratio but without steam addition. When the specific power was expressed in terms of compressor inlet air, the increase due to steam addition was larger. The average increase in this specific power at maximum efficiency was approximately 40 percent. The corresponding increase in specific power at the same pressure ratio was approximately 30 percent.

Cycle parameters. - Table V shows the cycle parameters for six cases. These cases are designated R1 to R6 to indicate recuperated cycle results. Five of these cases are for cycles with steam addition, and they were chosen using the data in figure 9 to give near maximum efficiency and high specific power. The other case, case R6, is included as a reference and is equivalent to case R4 without the evaporator. The evaporator flow rate ratio is the amount of water vaporized per unit mass of compressor inlet air flow. This rate is governed by the compressor exit temperature. The evaporator flow rate ratio was significantly less than the simple cycle steam addition flow rate ratio shown in table II for the corresponding turbine inlet temperature. Consequently, the performance gains to a recuperative cycle due to steam addition were less than the gains to a simple cycle due to steam addition.

Effect of turbine inlet temperature. - Figure 10 gives the cycle efficiency for recuperated cycles as a function of turbine inlet temperature. Data are given at various pressure ratios for cases with and without steam addition. For the cases with steam addition the air is assumed to exit the evaporator saturated. The data in figure 10(a) assume an uncooled turbine, and the data in figure 10(b) are for an air-cooled turbine. In contrast to the data for the simple cycle, these data show no discontinuity in slope with increasing turbine inlet temperature for the cycle with steam addition. The steam addition cases show the same performance trends as do the results for the recuperated cycle without steam addition, except that maximum efficiency occurs at higher pressure ratios.

Perturbation analysis. - The information in tables VI and VII gives the effect on the performance of the recuperated cycle due to variations in the parameters used in the analysis. The approach used to determine the influence coefficients was the same as that used for the data in tables III and IV for the simple cycle with steam addition. These data are given for each case shown in table V. The data in table VII show that cycle efficiency is sensitive to pressure losses. Consequently, the relative improvement in efficiency due to the addition of the evaporator is strongly dependent on the value assumed for the evaporator pressure drop. From table VII it can be seen that if the evaporator pressure drop increased from 4 to 6 percent the cycle efficiency would decrease half a percentage point for case R1. The specific power would decrease about 2 percent for the same case. Conversely, if the evaporator pressure drop were decreased from 4 to 2 percent, the efficiency and specific power would increase the same amounts.

When cases R4 and R6 are compared in table VII the influence of compressor polytropic efficiency on cycle efficiency is seen to be greater for the case without an evaporator. Cases R4 and R6 were both calculated at a pressure ratio of 12. When there is no evaporator present the maximum efficiency is achieved at a lower pressure ratio. The influence coefficient for compressor polytropic efficiency on cycle efficiency was calculated for the same case as R6 except for a pressure ratio of 6. This pressure ratio corresponded to near maximum cycle efficiency. Because of the lower pressure ratio, the influence coefficient was less. Its value of 0.53 was almost the same as the influence coefficient for case R4 with the evaporator present.

Table VII shows that the recuperator effectiveness significantly affects cycle efficiency but not specific power for the five cases with steam addition. For case R6,

the recuperator effectiveness influence coefficient of 0.12 is much less than the influence coefficients for the cases with steam addition. This is the result of case R6 being calculated for a pressure ratio greater than that for maximum efficiency. For case R6, with the pressure ratio changed to 6, the influence coefficient for recuperator effectiveness on cycle efficiency was 0.36. This coefficient value was slightly greater than the coeffcient value of 0.30 for case R4.

The influence coefficients shown in tables IV and VII show the effect of changing only one parameter at a time. In order to maintain maximum efficiency when one parameter is changed, other parameters may have to be changed. For example, table VII indicates that increasing the recuperator effectiveness from 0.85 to 0.90 for case R4 would increase the cycle efficiency by 1.5 percentage points. However, at a recuperator effectiveness of 90 percent, the maximum cycle efficiency occurs at a pressure ratio of 8. At these conditions the maximum cycle efficiency increased by 2.1 percentage points over the maximum efficiency shown in figure 9(d) for the cycle with steam addition.

Figure 8 shows that efficiency increases with the amount of water vaporized in the evaporator. However, the temperature at the outlet of the evaporator also changes with the amount of water vaporized. Therefore, the relative humidity of the air leaving the evaporator does not change as rapidly with the amount of water vaporized. Consequently, the influence of outlet relative humidity on cycle efficiency is small, as shown in table VII. The data in table VII indicate that at the same pressure ratio the cycle efficiency is improved between 2 and 4 percentage points due to the addition of an evaporator to provide steam.

Comparison of Simple and Recuperated Cycle Performance

Tables II and V and figures 5 and 9 can be used to make some observations regarding the relative performance of simple and recuperated cycles with steam addition. Figure 5 shows that at the same turbine inlet temperature and cooling assumption the maximum cycle efficiency for the simple cycle with steam addition was, at most, 4 percentage points greater than the maximum efficiency for the recuperated cycle without steam addition. The difference was even less for a recuperated cycle employing an evaporator. The calculations were done with the same assumptions for cooling requirements, pressure losses, and component efficiencies. However,

if the recuperator effectiveness was sufficiently increased, the recuperated cycle would have the higher efficiency. A recuperator effectiveness of 0.85 was chosen as typical based on past analyses. The recuperator effectiveness for a particular application is a function of both technical and economic factors.

Comparing the data in tables II and V shows that less steam is vaporized in the evaporator of the recuperated cycle than is generated in the boiler of the corresponding simple cycle. Approximately 30 percent less water was used for the same output in the recuperated cycle than in the corresponding simple cycle.

Tables IV and VII show that turbine cooling has a greater effect on the performance of the recuperated cycle than on the simple cycle. Consequently, the number of turbine stages used in the analysis has a greater effect for the recuperated cycle. For the simple cycle data in table IV, the amount of steam injected remained the same when the number of turbine stages varied. However, for the recuperated cycle data in table VII, the amount of steam formed in the evaporator decreased as the number of turbine stages increased, due to the greater amount of coolant required.

Comparison of Relative Heat Exchange Area

The difference in specific power at a point near maximum efficiency between the simple cycle with steam addition and either recuperated cycle is much greater than the differences in maximum efficiency. Increased specific power tends to result in decreased capital costs for the turbomachinery. It also has a beneficial effect on the size and, consequently, cost of the heat exchanger. For a given power level a greater specific power results in less flow through the heat exchanger. Table VIII compares the relative heat exchanger area per unit of output power for the recuperated cycles with and without steam addition and the simple cycle with steam addition. The heat exchanger for the simple cycle is the heat recovery boiler. The evaporator size was neglected for the recuperator cycle with steam addition. These data are only an indication of the relative areas. The required areas were calculated assuming a constant overall heat-transfer coefficient for all recuperated cases. The heat recovery boiler was divided into three sections - preheat, boiling, and superheat. The overall heat-transfer coefficient for the superheated section was assumed to be the same as that for the recuperated cycle. For the boiling and preheat sec-

tions the heat-transfer coefficient was taken to be twice the value for the superheated section. In table VIII the relative area for the recuperated cycle without an evaporator at 816° C (1500° F) was set equal to 1. All other areas are shown relative to this value. Data are given for each turbine inlet temperature and cooling assumption as was used in table II. For the recuperated cycle without an evaporator, the pressure ratios were chosen to give near maximum efficiency and high specific power (see fig. 9). Comparisons using tables II, V, and VIII show that, for the same turbine inlet temperature and cooling assumption, relative areas decreased as the amount of steam increased. This is primarily due to the increased specific power associated with the increased steam addition. Even though the mass flow through the recuperator increased with the addition of the evaporator, there was a greater increase in specific power resulting in a smaller relative area. The relative area for the simple cycle with the heat recovery boiler was even less. The relative area per unit flow was less because of the large inlet temperature differences between the fluids in the boiler and the higher heat-transfer coefficient. The relative area per unit of output power is further decreased because of the greater specific power of the simple cycle with steam addition.

Water Recovery

The question of whether the water used for steam addition would be recovered for reuse would be decided on economic and environmental considerations. The performance results presented are largely independent of whether or not water is recovered from the exhaust stream. If water was recovered, there would be additional pressure losses and auxilary power requirements, but these are expected to be small. From a thermodynamic standpoint, it is relatively easy to recover the water used to generate steam for either the simple or recuperated cycles. Condensation of the water vapor will begin when the temperature of the exhaust gases is below the dew point temperature. All the water used to generate steam will condense if the temperature of the exhaust gases is sufficiently reduced. Figure 11 shows the temperature to which the exhaust gases must be cooled in order to get zero net water loss for each of the twelve cases shown in tables II and V. For the same turbine inlet temperature and cooling assumption there was less steam added for the recuperated cases. Consequently, the recuperated cycle data are in a nar-

rower range of condensation temperatures. For clarity the discussion will focus on the simple cycle cases. Case S4 differs from case S6 in that the later has no steam addition. For case S6, there is no water condensed in cooling the gases to this temperature. The location of case S6 in figure 11 shows the temperature at which the water formed by combustion is just beginning to condense. The higher condensation temperature in this figure for case S4 compared to case S6 illustrates the effect of additional water vapor created as a result of the greater fuel to air ratio. As expected, the higher the turbine inlet temperature, the higher the temperature at which all of the steam can be condensed. This is the result of more fuel per unit of air for the higher turbine inlet temperatures.

In figure 12 the water use rate as a function of the temperature to which the exhaust gases are cooled is shown for case S4. For temperatures higher than 66° C (150° F), no water would be condensed. However, the water use rate would not increase, since the water use rate would equal the steam generated in the heat recovery boiler. In agreement with figure 11, if the exhaust gas temperature is reduced to 36° C (97° F), the net consumption of water is zero. The shape of the curve shows that partial water recovery may be economically feasible, even if full water recovery is not. Half of the water could be recovered at 57° C (135° F), which is more than midway between the temperature for full recovery and the temperature for no recovery.

It does not seem feasible to use wet cooling towers to cool the exhaust gas for water recovery. With a wet cooling tower it is likely that significantly more water would be consumed in the cooling tower than would be recovered from the exhaust. In addition to the heat load resulting from vapor condensation, there is the heat load resulting from having to cool all of the exhaust to the condensation temperature. For example, to cool all the exhaust gas to 36° C (97° F) for case S4 imposes an additional heat load of approximately 35 percent. For a wet cooling tower to absorb both heat loads would result in a theoretical water consumption rate of 135 percent of the rate of vapor condensed. Wet cooling towers have additional drift and blowdown losses which would increase the water consumption rate even further.

Even with dry cooling towers there are additional factors to consider when choosing whether or not to recover the water vapor from the exhaust gas. When water is recovered, gas which leaves the water recovery unit would be saturated and at a low temperature. This saturated vapor would likely contain corrosive

acids. To minimize corrosion problems in the exhaust stack, the gas may have to be reheated. This could be done with an additional heat exchanger and its associated capital costs or with an additional burner which would reduce plant efficiency.

CONCLUDING REMARKS

The results given here, and in previous work, show that steam addition can be an effective way of improving the thermodynamic performance of simple and recuperated gas turbine cycles. For a simple cycle, significant improvements in both cycle efficiency and specific power can be achieved. For the cases examined, the maximum simple cycle efficiency increased between 25 and 30 percent as a result of steam addition. The increase in specific power at maximum efficiency for the simple cycle with steam addition was between 25 and 35 percent. The increase was calculated at the pressure ratio corresponding to maximum efficiency for the case with steam addition. For a recuperated cycle, steam addition resulted in smaller increases in cycle efficiency but significant increases in specific power. The maximum cycle efficiency increased approximately 3 percent (1 percentage point). The specific power increased between 10 and 25 percent with an average of 20 percent, as a result of steam addition. At maximum efficiency the steam addition rate for a simple cycle is considerably greater than that for a recuperated cycle. Consequently, the specific power at maximum efficiency is greater for the simple cycle.

The simple cycle pressure ratio required for maximum efficiency is less with steam addition than without steam addition at the same turbine inlet temperature. Therefore, there is less incentive to increase the compressor pressure ratio with increased turbine inlet temperature for a cycle with steam addition. In the recuperated cycle with steam addition there is less of a penalty for using pressure ratios different from optimum than in a recuperated cycle without steam addition. For the uncooled turbine assumption the maximum cycle efficiency increased markedly with turbine inlet temperature. This was not true for the air-cooled turbine. The small increase in cycle efficiency between an air-cooled case at 1093° C (2000° F) and an air-cooled case at 1371° C (2500° F) is a result of the large increase in cooling requirements at the higher turbine inlet temperature. One way of increasing the efficiency of a cooled turbine is to provide a lower temperature

coolant. Only two cycles have been examined in this report to illustrate the benefits of steam addition. Other Brayton cycle variations may offer greater performance improvements and/or be more economical. For example, the use of a compressor intercooler to simultaneously preheat the water into the heat recovery boiler and lower the compressor discharge temperature may be beneficial. The intercooler itself is beneficial in that it reduces compressor work. The required turbine coolant flow is also reduced since the coolant is at a lower temperature. As a second example, the use of evaporator discharge air in place of the compressor discharge air for turbine cooling may improve the cycle performance.

The water use rate for the simple cycle with steam addition was less than half of that for a conventional coal-fired steam plant employing wet cooling towers. For cases examined near the maximum cycle efficiency, the water use rate was between 1.06 and 1.67 kilograms per kilowatt-hour (0.28 and 0.44 gal/kW-hr). The water use rate for the recuperated cycle with steam addition was even less. From a thermodynamic standpoint, it is relatively easy to recover the water used for steam injections. However, when cost and corrosion problems are taken into account, it may not be desirable to do so.

The required heat-transfer area per unit of output power was less for the simple cycle with steam addition than for the recuperator at the same turbine inlet temperature. Consequently, the cost of the heat recovery boiler would probably be less than the cost of a recuperator for the same output power.

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TABLE I. - ASSUMED VALUES USED IN THE ANALYSIS

Description	Nominal value or range
Simple cycle with steam addition:	
Steam flow rate mass ratio, percent	0 to 40
Turbine inlet temperature, K (^{O}R)	1089 to 1644 (1960 to 2960)
Compressor pressure ratio	6 to 24 as appropriate
Compressor polytropic efficiency, percent	87
Turbine polytropic efficiency, percent	87
Turbine mechanical efficiency, percent	98
Generator efficiency, percent	98
Compressor inlet temperature, K (^{O}R)	288 (519)
Compressor inlet pressure, N/cm^2 (psia)	10.0 (14.5)
Exhaust outlet pressure, N/cm^2 (psia)	10.1 (14.7)
Exhaust pressure drop in boiler, percent	4
Burner efficiency, percent	99
Burner pressure drop, percent	4
Inlet humidity, percent	60
Hydrogen to carbon ratio in fuel	1.942
Fuel heating value, MJ/kg (Btu/lb)	43.2 (18 600)
Minimum, ΔT, ^{O}C (^{O}F)	28 (50)
Steam pressure drop in boiler, percent	12
Water inlet temperature, K (^{O}R)	288 (519)
Pump efficiency, percent	70
Maximum metal temperature, K (^{O}R)	1089 (1960)
Blockage coolant ratio, (percent/stage)[a]	0.5
Additional values for recuperated cycle:	
Evaporator relative humidity, percent	100
Evaporator pressure drop, percent	4
Recuperator effectiveness, percent	85
Recuperator air side pressure drop, percent	2
Recuperator exhaust side pressure drop, percent	4

[a]Percent of turbine inlet flow per turbine stage.

TABLE II. - PERFORMANCE OF SELECTED CASES FOR SIMPLE CYCLE WITH STEAM ADDITION

Parameter	Case					
	S1	S2	S3	S4	S5	S6
	Reference figure					
	5(a)	5(b)	5(c)	5(d)	5(e)	5(d) (no steam)
Cycle efficiency (LHV), percent	33.6	41.2	42.4	43.0	49.7	33.3
Specific power, kW-hr/kg (Btu/lb):						
Per unit compressor air	0.064 (99.1)	0.116 (180)	0.160 (248)	0.131 (203)	0.227 (352)	0.078 (121)
Per unit of equivalent turbine air	0.054 (83.2)	0.098 (151)	0.145 (224)	0.101 (156)	0.158 (245)	0.076 (118)
Water use rate, kg/kW-hr (gal/kW-hr)	1.67 (0.44)	1.36 (0.36)	1.48 (0.39)	1.29 (0.34)	1.06 (0.28)	0.0
Compressor:						
Pressure ratio	10	16	16	16	24	16
Inlet:						
Temperature, ^{o}C (^{o}F)	15 (59)	15 (59)	15 (59)	15 (59)	15 (59)	15 (59)
Pressure, N/cm^2 (psia)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)
Relative humidity, percent	60	60	60	60	60	60
Outlet:						
Temperature, ^{o}C (^{o}F)	328 (623)	420 (788)	420 (788)	420 (788)	508 (946)	420 (788)
Pressure, N/cm^2 (psia)	100 (145)	160 (232)	160 (232)	160 (232)	240 (348)	160 (232)
Turbine:						
Pressure ratio	9.1	14.6	14.6	14.6	21.9	15.2
Inlet:						
Temperature, ^{o}C (^{o}F)	816 (1500)	1093 (2000)	1371 (2500)	1093 (2000)	1371 (2500)	1093 (2000)
Pressure, N/cm^2 (psia)	96 (139)	154 (223)	154 (223)	154 (223)	230 (334)	154 (223)
Inlet mass flow rate ratio	1.122	1.097	0.960	1.196	1.282	1.020
Coolant mass flow rate ratio	0.0	0.084	0.308	0.0	0.0	0.0
Outlet:						
Temperature, ^{o}C (^{o}F)	412 (773)	513 (956)	643 (1189)	534 (993)	657 (1214)	503 (937)
Pressure, N/cm^2 (psia)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.1 (14.7)
Boiler:						
Exhaust inlet:						
Temperature, ^{o}C (^{o}F)	412 (773)	513 (956)	643 (1189)	534 (993)	657 (1214)	----------
Pressure, N/cm^2 (psia)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	----------
Mass flow rate ratio	1.122	1.181	1.268	1.196	1.282	----------
Exhaust outlet:						
Temperature, ^{o}C (^{o}F)	154 (309)	146 (294)	118 (244)	141 (285)	123 (254)	----------
Pressure, N/cm^2 (psia)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	----------
Water inlet:						
Temperature, ^{o}C (^{o}F)	15 (59)	15 (59)	15 (59)	15 (59)	15 (59)	----------
Pressure, N/cm^2 (psia)	112 (162)	179 (260)	179 (260)	179 (260)	269 (390)	----------
Mass flow rate ratio	0.106	0.157	0.236	0.170	0.244	----------
Steam outlet:						
Temperature, ^{o}C (^{o}F)	384 (723)	486 (906)	615 (1139)	504 (940)	629 (1164)	----------
Pressure, N/cm^2 (psia)	100 (145)	160 (232)	160 (232)	160 (232)	240 (348)	----------
Burner:						
Steam mass flow rate ratio	0.106	0.157	0.236	0.170	0.244	0.0
Air mass flow rate ratio	1.000	0.916	0.692	1.000	1.000	1.000
Fuel mass flow rate ratio	0.016	0.024	0.032	0.026	0.038	0.020
Excess air, percent	327	164	49	177	78	248

TABLE III. - DESCRIPTION OF INFLUENCE COEFFICIENTS FOR SIMPLE CYCLE WITH STEAM ADDITION

Parameter number	Parameter description	Definition of X^a	Value of x^{*b}
1	Steam flow rate mass ratio, percent	$X = x - x^*$	10.6 - case 1 15.7 - case 2 23.6 - case 3 17.0 - case 4 24.4 - case 5 0.0 - case 6
2	Turbine inlet temperature, K ($^{\circ}$R)	$X = 100\ (x - x^*)/x^*$	1089 (1960) - case 1 1367 (2460) - cases 2, 4, and 5 1644 (2960) - cases 3 and 5
3	Compressor pressure ratio	$X = 100\ (x - x^*)/x^*$	10 - case 1 16 - cases 2, 3, 4, and 6 24 - case 5
4	Compressor polytropic efficiency, percent	$X = x - x^*$	87 - all cases
5	Turbine polytropic efficiency, percent	$X = x - x^*$	87
6	Turbine mechanical efficiency, percent	$X = x - x^*$	98
7	Generator efficiency, percent	$X = x - x^*$	98
8	Compressor inlet temperature, K ($^{\circ}$R)	$X = 100\ (x - x^*)/x^*$	288 (519)
9	Compressor inlet pressure, N/cm^2 (psia)	$X = 100\ (x - x^*)/x^*$	10.0 (14.5)
10	Exhaust outlet pressure, N/cm^2 (psia)	$X = 100\ (x - x^*)/x^*$	10.1 (14.7) - all cases
11	Exhaust pressure drop in boiler, percent	$X = x - x^*$	4
12	Burner efficiency, percent	$X = x - x^*$	99
13	Burner pressure drop, percent	$X = x - x^*$	4
14	Inlet humidity, percent	$X = x - x^*$	60
15	Hydrogen to carbon ratio in fuel	$X = 100\ (x - x^*)/x^*$	1.942 - all cases
16	Fuel heating value, MJ/kg (Btu/lb)	$X = 100\ (x - x^*)/x^*$	43.2 (18 600)
17	Minimum ΔT, $^{\circ}$C ($^{\circ}$F)	$X = 100\ (x - x^*)/x^*$	28 (50)
18	Steam pressure drop in boiler, percent	$X = x - x^*$	12
19	Water inlet temperature, K ($^{\circ}$R)	$X = 100\ (x - x^*)/x^*$	288 (519)
20	Pump efficiency, percent	$X = x - x^*$	70 - all cases
21	Any cooling	$X = 1$ if $x \neq x^*$	0 - cases 1, 4, 5, and 6 1 - cases 2 and 3
22	Maximum metal temperature, K ($^{\circ}$R)	$X = 100\ (x - x^*)/x^*$	1089 (1960) - cases 2, 3, 4, and 6 1144 (2060) - case 5
23	Blockage coolant ratio, percent/stage	$X = x - x^*$	0.5 - all cases
24	Number of turbine stages	$X = x - x^*$	4 - case 2 5 - cases 4 and 6 6 - cases 3 and 5

[a]Normalized independent variable.

[b]Value of independent variable at base case.

TABLE IV. - EFFICIENCY AND SPECIFIC POWER INFLUENCE COEFFICIENTS FOR SIMPLE CYCLE BASE CASES

Parameter number	Parameter description	Case											
		S1		S2		S3		S4		S5		S6	
		Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$
1	Steam flow rate mass ratio,												
	$X > 0$	-0.28	2.04	-0.16	1.45	-0.22	0.92	-0.24	1.25	-0.19	0.84	-----	-----
	$X < 0$	.67	2.09	.54	1.43	.37	.94	.50	1.25	.38	.83	-----	-----
2	Turbine inlet temperature,												
	$X > 0$	.11	2.55	-.11	1.81	-.13	1.50	.03	2.05	-.03	1.79	0.15	2.71
	$X < 0$	.56	2.51	.37	1.99	.12	1.52	.50	2.02	.43	1.77	.18	2.70
3	Compressor pressure ratio,												
	$X > 0$	-.03	-.04	-.01	.03	-.02	.14	-.01	.04	.01	.07	.05	-.14
	$X < 0$	.10	.05	.09	.06	.08	.16	.10	.07	.09	.08	.05	-.12
4	Compressor polytropic efficiency	0.49	2.24	0.41	1.68	0.37	1.16	0.36	1.52	0.27	1.13	0.55	2.55
5	Turbine polytropic efficiency	.58	2.14	.55	1.75	.48	1.60	.50	1.62	.44	1.36	.69	2.08
6	Turbine mechanical efficiency	.82	2.43	.84	2.03	.74	1.75	.83	1.92	.82	1.66	.85	2.53
7	Generator efficiency	.34	1.02	.42	1.02	.43	1.02	.44	1.02	.51	1.02	.34	1.02
8	Compressor inlet temperature	-.16	-1.33	-.14	-.93	-.15	-.65	-.10	-.84	-.07	-.59	-.20	-1.38
9	Compressor inlet pressure	.22	.83	.18	.57	.15	.52	.16	.52	.12	.38	.22	.66
10	Exhaust outlet pressure	-0.24	-0.84	-0.21	-0.57	-0.14	-0.51	-0.17	-0.53	-0.13	-0.39	-0.22	-0.67
11	Exhaust pressure drop in boiler	-.23	-.80	-.19	-.54	-.15	-.50	-.15	-.50	-.12	-.37	-----	-----
12	Burner efficiency	.34	-.04	.42	-.04	.42	-.03	.44	-.04	.51	-.04	.34	-.06
13	Burner pressure drop	-.25	-.88	-.20	-.59	-.16	-.54	-.18	-.55	-.14	-.40	-.23	-.70
14	Inlet humidity	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)
15	Hydrogen to carbon ratio in fuel	(a)	0.02	(a)	0.02	(a)	0.02	(a)	0.02	(a)	0.02	(a)	0.03
16	Fuel heating value	(a)	-.03	(a)	-.04	(a)	-.03	(a)	-.03	(a)	-.04	(a)	-.06
17	Minimum ΔT	-0.01	.001	-0.01	.001	-0.01	.001	-0.01	.001	-0.01	(a)	-----	-----
18	Steam pressure drop in boiler	(a)	(a)	(a)	(a)	(a)	-.001	-.003	(a)	-.003	(a)	-----	-----
19	Water inlet temperature	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	-----	-----
20	Pump efficiency	0.001	0.003	0.002	0.004	0.002	0.004	0.002	0.004	0.002	0.005	-----	-----
21	Any cooling	------	------	1.09	1.78	3.28	3.94	-1.97	-1.52	-5.61	-2.24	-1.53	-4.63
22	Maximum metal temperature	------	------	.10	.12	.32	.25	.16	.11	.39	.17	.11	.35
23	Blockage coolant ratio, percent/stage	------	------	-1.47	-3.43	-1.56	-2.87	-1.55	-3.21	-1.80	-3.29	-2.00	-6.31
24	Number of turbine stages	------	------	-.35	-.63	-.48	-.58	-.46	-.59	-.70	-.44	-.49	-1.37

[a]Influence coefficient approximately zero.

TABLE V. - PERFORMANCE OF SELECTED CASES FOR RECUPERATED CYCLE WITH STEAM ADDITION

Parameter description	Case					
	R1	R2	R3	R4	R5	R6
	Reference figure					
	9(a)	9(b)	9(c)	9(d)	9(e)	9(d) (no steam)
Cycle efficiency (LHV), percent	33.8	39.6	41.0	41.7	47.0	38.1
Specific power, kW-hr/kg (Btu/lb):						
Per unit compressor air	0.054 (83.1)	0.093 (144)	0.117 (181)	0.101 (157)	0.160 (247)	0.075 (116)
Per unit of equivalent turbine air	0.047 (72.9)	0.084 (130)	0.112 (174)	0.086 (133)	0.130 (202)	0.074 (114)
Water use rate, kg/kW-hr (gal/kW-hr)	1.40 (0.37)	0.98 (0.26)	0.72 (0.19)	0.98 (0.26)	0.72 (0.19)	0.0
Compressor:						
Pressure ratio	8	12	12	12	16	12
Inlet:						
Temperature, ^{o}C (^{o}F)	15 (59)	15 (59)	15 (59)	15 (59)	15 (59)	15 (59)
Pressure, N/cm^2 (psia)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)	10.0 (14.5)
Relative humidity, percent	60	60	60	60	60	60
Outlet:						
Temperature, ^{o}C (^{o}F)	289 (552)	363 (685)	363 (685)	363 (685)	420 (788)	363 (685)
Pressure, N/cm^2 (psia)	80 (116)	120 (174)	120 (174)	120 (174)	160 (232)	120 (174)
Evaporator:						
Mass flow rate ratio	0.076	0.093	0.084	0.098	0.116	----------
Outlet:						
Temperature, ^{o}C (^{o}F)	97 (207)	113 (236)	111 (232)	115 (239)	128 (263)	----------
Pressure, N/cm^2 (psia)	77 (111)	115 (167)	115 (167)	115 (167)	154 (223)	----------
Turbine:						
Pressure ratio	6.85	10.3	10.3	10.3	13.7	10.7
Inlet:						
Temperature, ^{o}C (^{o}F)	816 (1500)	1093 (2000)	1371 (2500)	1093 (2000)	1371 (2500)	1093 (2000)
Pressure, N/cm^2 (psia)	72 (105)	108 (157)	108 (157)	108 (157)	145 (210)	113 (164)
Inlet mass flow rate ratio	1.089	1.053	0.949	1.119	1.144	1.016
Coolant mass flow rate ratio	0.0	0.060	0.159	0.0	0.0	0.0
Outlet:						
Temperature, ^{o}C (^{o}F)	452 (845)	564 (1048)	707 (1305)	584 (1083)	724 (1336)	561 (1042)
Pressure, N/cm^2 (psia)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)
Recuperator:						
Exhaust inlet:						
Temperature, ^{o}C (^{o}F)	452 (845)	564 (1048)	707 (1305)	584 (1083)	724 (1336)	561 (1042)
Pressure, N/cm^2 (psia)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)	10.5 (15.3)
Mass flow rate ratio	1.089	1.113	1.108	1.119	1.144	1.016
Exhaust outlet:						
Temperature, ^{o}C (^{o}F)	163 (325)	222 (432)	304 (580)	209 (408)	258 (496)	401 (754)
Pressure, N/cm^2 (psia)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)	10.1 (14.7)
Air inlet:						
Temperature, ^{o}C (^{o}F)	97 (207)	113 (236)	111 (232)	115 (239)	128 (263)	363 (685)
Pressure, N/cm^2 (psia)	77 (111)	115 (167)	115 (167)	115 (167)	154 (223)	120 (174)
Mass flow rate ratio	1.076	1.033	0.926	1.098	1.116	1.000
Air outlet:						
Temperature, ^{o}C (^{o}F)	398 (749)	497 (927)	618 (1144)	513 (956)	635 (1175)	531 (988)
Pressure, N/cm^2 (psia)	75 (109)	113 (164)	113 (164)	113 (164)	150 (218)	118 (171)
Burner:						
Airstream mass flow rate ratio	1.076	1.033	0.926	1.098	1.116	1.000
Fuel mass flow rate ratio	0.013	0.020	0.024	0.020	0.028	0.016
Excess air, percent	412	225	142	234	140	313

TABLE VI. - DESCRIPTION OF INFLUENCE COEFFICIENTS FOR RECUPERATED CYCLE WITH STEAM ADDITION

Parameter number	Parameter description	Definition of X[a]	Value of x^{*b}
2	Turbine inlet temperature, K ($^{\circ}$R)	$X = 100 (x - x^*)/x^*$	1089 (1960) - case 1 1367 (2460) - cases 2, 4, and 6 1644 (2960) - cases 3 and 5
3	Compressor pressure ratio	$X = 100 (x - x^*)/x^*$	8 - case 1 12 - cases 2, 3, 4, and 6 16 - case 5
25	No evaporator	$X = 1$ if $x \neq x^*$	1 - cases 1 to 5
26	Evaporator relative humidity, percent	$X = x - x^*$	100 - all cases
27	Evaporator pressure drop, percent	$X = x - x^*$	4 - all cases
4	Compressor polytropic efficiency, percent	$X = x - x^*$	87 - all cases
5	Turbine polytropic efficiency, percent	$X = x - x^*$	87
6	Turbine mechanical efficiency, percent	$X = x - x^*$	98
7	Generator efficiency, percent	$X = x - x^*$	98
8	Compressor inlet temperature, K ($^{\circ}$R)	$X = 100 (x - x^*)/x^*$	288 (519)
9	Compressor inlet pressure, N/cm^2 (psia)	$X = 100 (x - x^*)/x^*$	10.0 (14.5)
28	Recuperator effectiveness, percent	$X = x - x^*$	85 - all cases
10	Exhaust outlet pressure, N/cm^2 (psia)	$X = 100 (x - x^*)/x^*$	10.1 (14.7)
29	Recuperator air side pressure drop, percent	$X = x - x^*$	2
30	Recuperator exhaust side pressure drop, percent	$X = x - x^*$	4
12	Burner efficiency, percent	$X = x - x^*$	99
13	Burner pressure drop, percent	$X = x - x^*$	4
14	Inlet humidity, percent	$X = x - x^*$	60 - all cases
15	Hydrogen to carbon ratio in fuel	$X = 100 (x - x^*)/x^*$	1.942
16	Fuel heating value, MJ/kg (Btu/lb)	$X = 100 (x - x^*)/x^*$	43.2 (18 600)
19	Water inlet temperature, K ($^{\circ}$R)	$X = 100 (x - x^*)/x^*$	288 (519)
20	Pump efficiency, percent	$X = x - x^*$	70
21	Any cooling	$X = 1$ if $x \neq x^*$	0 - cases 1, 4, 5, and 6 1 - cases 2 and 3
22	Maximum metal temperature, K ($^{\circ}$R)	$X = 100 (x - x^*)/x^*$	1089 (1960) - cases 2, 3, 4, and 6 1144 (2060) - case 5
23	Blockage coolant ratio, percent/stage	$X = x - x^*$	0.5 - all cases
24	Number of turbine stages	$X = x - x^*$	3 - cases 2, 3, 4, and 6 4 - case 5

[a]Normalized independent variable.

[b]Value of independent variable at base case.

TABLE VII. - EFFICIENCY AND SPECIFIC POWER INFLUENCE COEFFICIENTS FOR RECUPERATED CYCLE BASE CASES

Parameter number	Parameter description	Case											
		R1		R2		R3		R4		R5		R6	
		Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$	Efficiency, $\Delta Y_1/\Delta X$	Specific power, $\Delta Y_2/\Delta X$
2	Turbine inlet temperature,												
	X > 0	0.41	2.55	0.18	1.96	-0.05	1.55	0.33	2.09	0.25	1.85	0.42	2.45
	X < 0	.47	2.54	.23	1.97	-.03	1.56	.37	2.08	.28	1.84	.46	2.44
3	Compressor pressure ratio,												
	X > 0	-.03	.17	-.02	.17	.01	.22	-.02	.18	-.01	.18	-.08	.02
	X < 0	-.03	.24	-.01	.21	.01	.24	-.02	.21	-.01	.20	-.08	.02
25	No evaporator	-3.10	-14.8	-3.65	-14.9	-2.06	-10.3	-3.61	-14.4	-3.52	-13.5	-----	-----
26	Evaporator relative humidity	.005	.02	.004	.02	.002	.01	.004	.02	.003	.01	-----	-----
27	Evaporator pressure drop	-.25	-1.07	-.20	-.73	-.20	-.69	-.19	-.69	-.15	-.53	-----	-----
4	Compressor polytropic efficiency	0.64	1.90	0.57	1.42	0.52	1.19	0.54	1.28	0.46	0.95	0.77	2.14
5	Turbine polytropic efficiency	.54	2.26	.51	1.88	.53	1.76	.47	1.77	.43	1.52	.52	2.05
6	Turbine mechanical efficiency	.83	2.48	.83	2.09	.78	1.88	.83	2.01	.83	1.76	.90	2.35
7	Generator efficiency	.34	1.02	.40	1.02	.42	1.02	.42	1.02	.48	1.02	.39	1.02
8	Compressor inlet temperature	-.33	-.96	-.28	-.67	-.28	-.59	-.26	-.59	-.21	-.40	-.41	-1.20
9	Compressor inlet pressure	.24	1.00	.18	.68	.19	.64	.17	.64	.14	.49	.19	.75
28	Recuperator effectiveness	0.26	-0.02	0.27	-0.02	0.28	-0.02	0.30	-0.02	0.33	-0.03	0.12	-0.01
10	Exhaust outlet pressure	-.24	-1.02	-.19	-.70	-.20	-.66	-.18	-.66	-.14	-.51	-.19	-.75
29	Recuperator air side pressure drop	-.25	-1.05	-.19	-.72	-.20	-.67	-.18	-.68	-.14	-.52	-.20	-.77
30	Recuperator exhaust side pressure drop	-.23	-.98	-.18	-.67	-.19	-.63	-.17	-.63	-.14	-.49	-.18	-.72
12	Burner efficiency	.34	-.03	.40	-.03	.43	-.03	.42	-.03	.46	-.04	.39	-.04
13	Burner pressure drop	-.26	-1.07	-.20	-.74	-.21	-.69	-.19	-.69	-.15	-.53	-.20	-.79
14	Inlet humidity	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)	(a)
15	Hydrogen to carbon ratio in fuel	0.002	0.02	0.002	0.02	0.001	0.02	0.002	0.02	(a)	0.02	(a)	0.02
16	Fuel heating value	(a)	-.03	.002	-.03	.011	-.03	.003	-.03	0.008	-.04	(a)	-.04
19	Water inlet temperature	.02	.07	.02	.06	.02	.05	.02	.06	.02	.05	-----	-----
20	Pump efficiency	.001	.002	.001	.002	.001	.002	.001	.002	.001	.002	-----	-----
21	Any cooling	------	-------	2.12	2.64	6.00	5.92	-2.12	-2.57	-7.24	-7.97	-2.16	-3.07
22	Maximum metal temperature	------	-------	.17	.22	.35	.43	.17	.22	.42	.61	.17	.25
23	Blockage coolant ratio, percent/stage	------	-------	-1.55	-3.25	-1.31	-2.48	-1.55	-3.25	-1.63	-3.49	-1.70	-4.09
24	Number of turbine stages	------	-------	-.65	-1.06	-1.37	-1.75	-.65	-1.06	-1.28	-1.97	-.64	-1.23

[a]Influence coefficient approximately zero.

TABLE VIII. - RELATIVE HEAT EXCHANGER AREA PER UNIT OF OUTPUT POWER

Turbine inlet temperature		Turbine cooling	Recuperated cycle				Simple cycle with heat recovery boiler	
			No evaporator		With evaporator			
°C	°F		Pressure ratio	Recuperator relative area	Pressure ratio	Recuperator relative area	Pressure ratio	Boiler relative area
816	1500	Uncooled	6	[a]1	8	0.82	10	0.26
1093	2000	Air cooled	8	.44	12	.41	16	.21
1371	2500	Air cooled	10	.30	12	.25	16	.20
1093	2000	Uncooled	8	.52	12	.44	16	.19
1371	2500	Uncooled	10	.34	16	.28	24	.18

[a]Relative area chosen as 1 for this case.

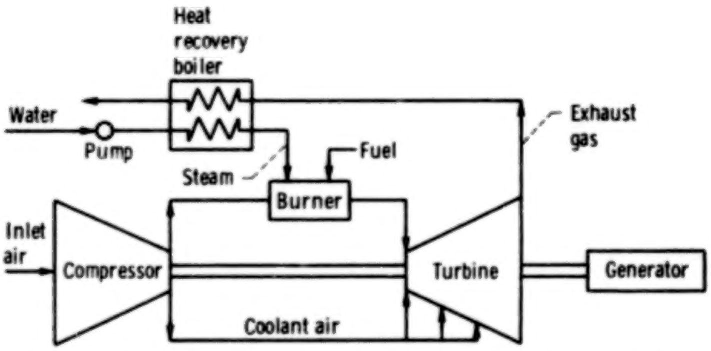


Figure 1. - Schematic diagram of simple gas turbine with heat recovery boiler.

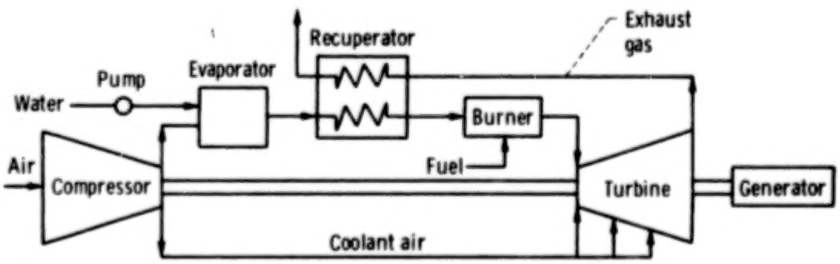
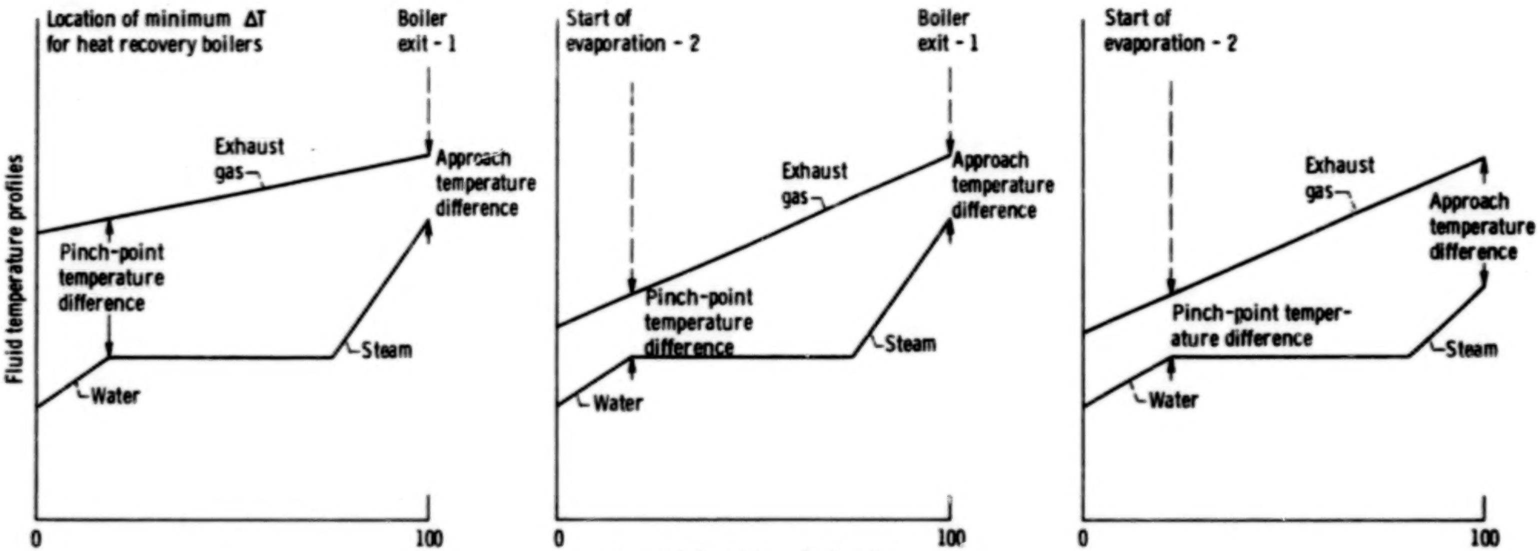


Figure 2. - Schematic diagram of recuperated gas turbine with evaporator in compressor discharge flow path.



(a) Minimum ΔT at steam exit of boiler.

(b) Minimum ΔT at both steam exit of boiler and pinch point within boiler.

(c) Minimum ΔT at pinch point within boiler.

Figure 3. - Schematic diagram of fluid temperature profiles in heat recovery boiler.

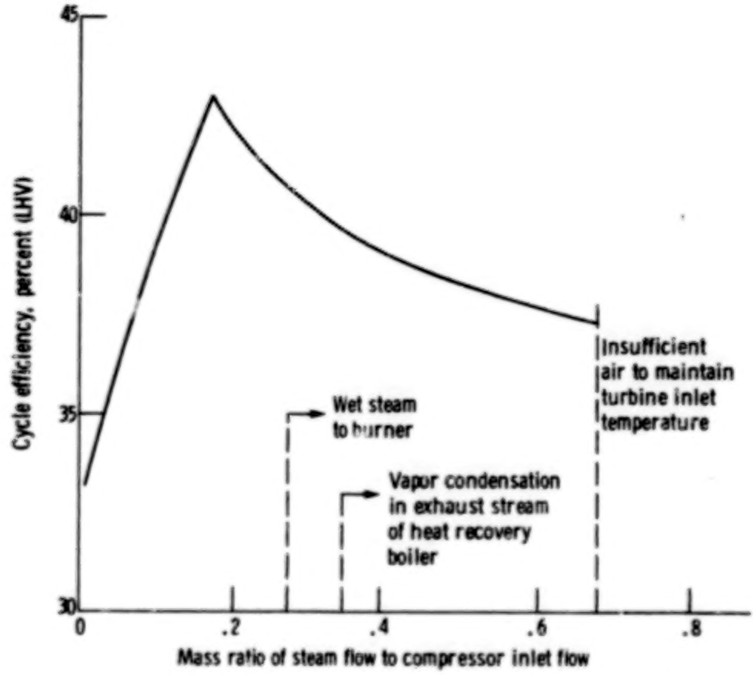
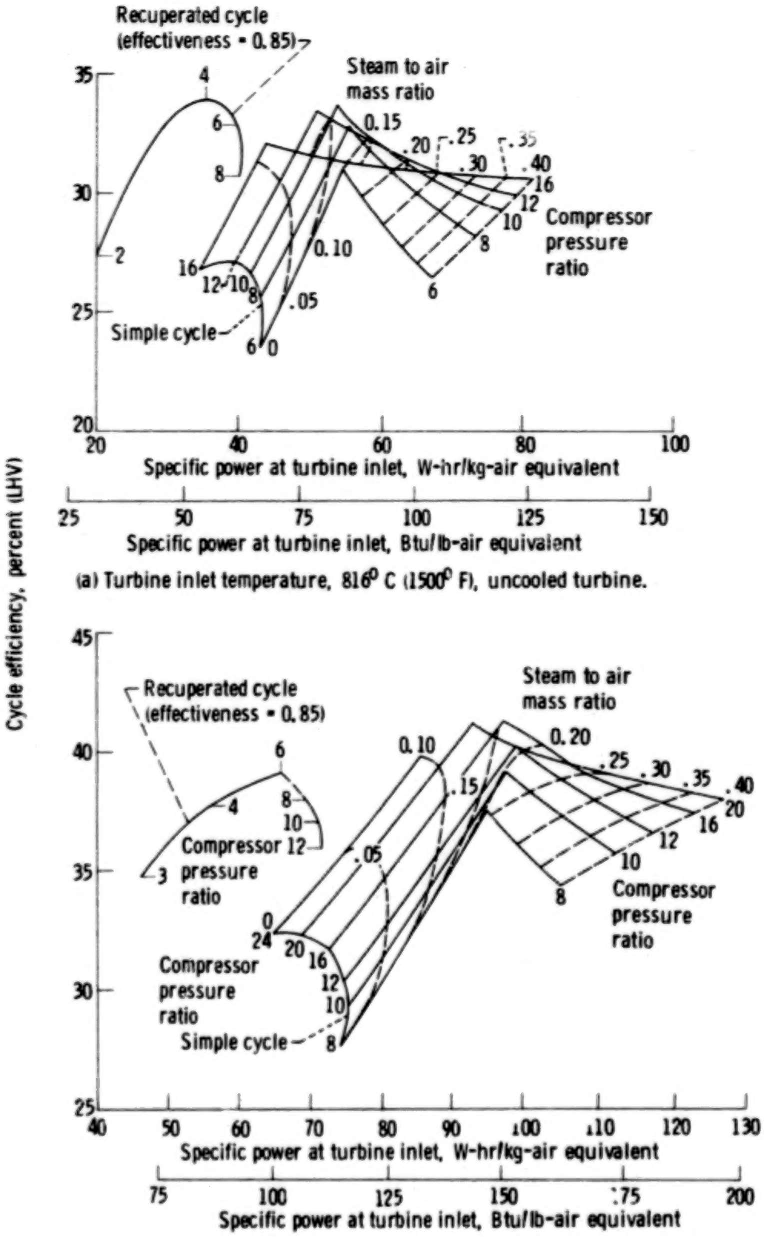


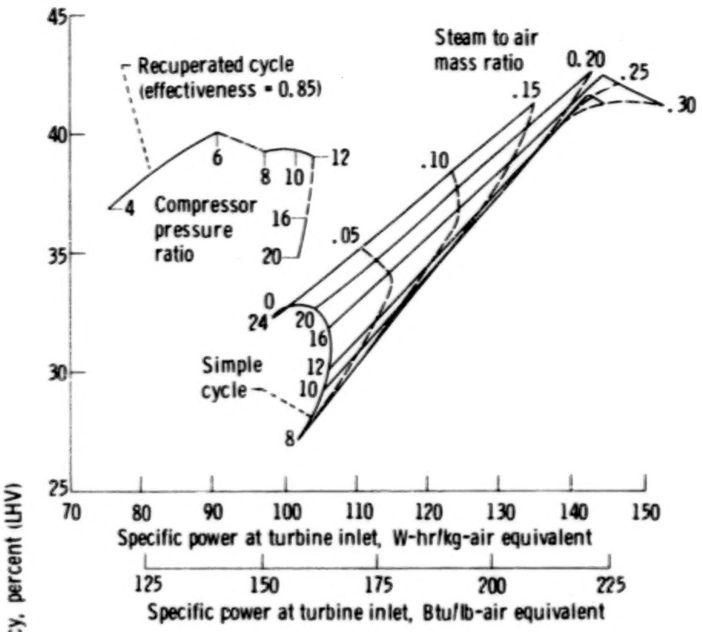
Figure 4. - Cycle efficiency as function of amount of steam generated in heat recovery boiler. Turbine inlet temperature, 1093° C (2000° F); uncooled turbine; compressor pressure ratio, 16.



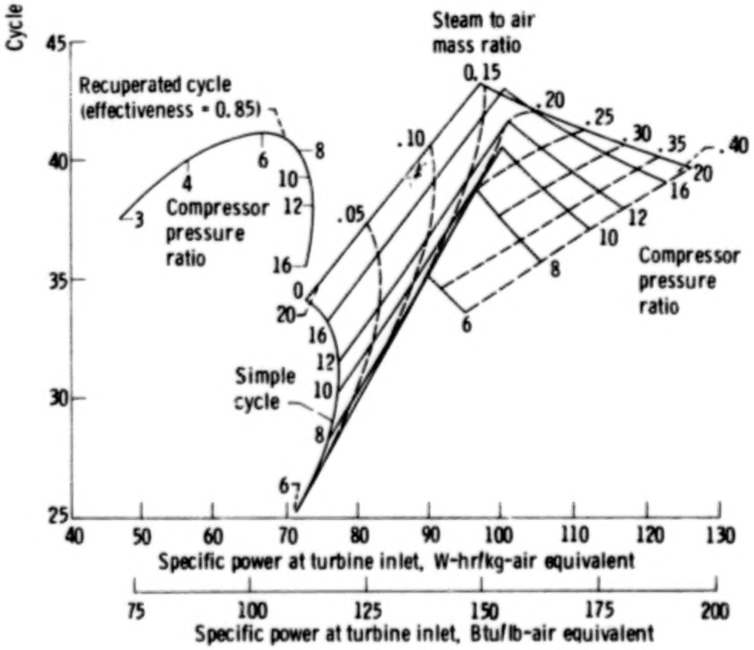
(a) Turbine inlet temperature, 816° C (1500° F), uncooled turbine.

(b) Turbine inlet temperature, 1093° C (2000° F); air-cooled turbine.

Figure 5. - Cycle efficiency and specific power for simple cycle with steam addition.

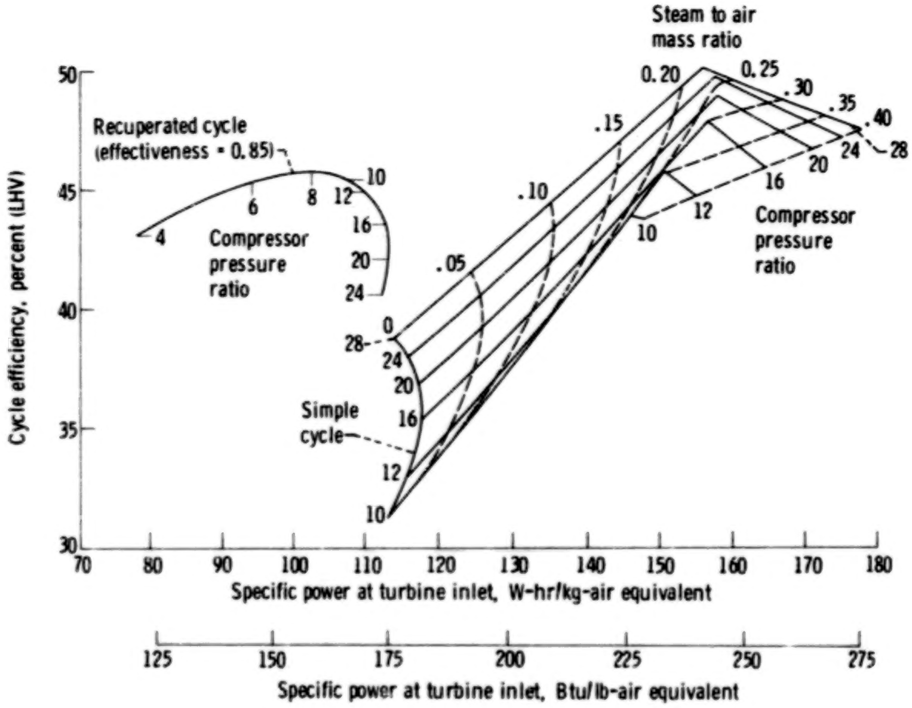


(c) Turbine inlet temperature, 1371° C (2500° F); air-cooled turbine.



(d) Turbine inlet temperature, 1093° C (2000° F), uncooled turbine.

Figure 5. - Continued.



(e) Turbine inlet temperature, 1371° C (2500° F), uncooled turbine.

Figure 5. - Concluded.

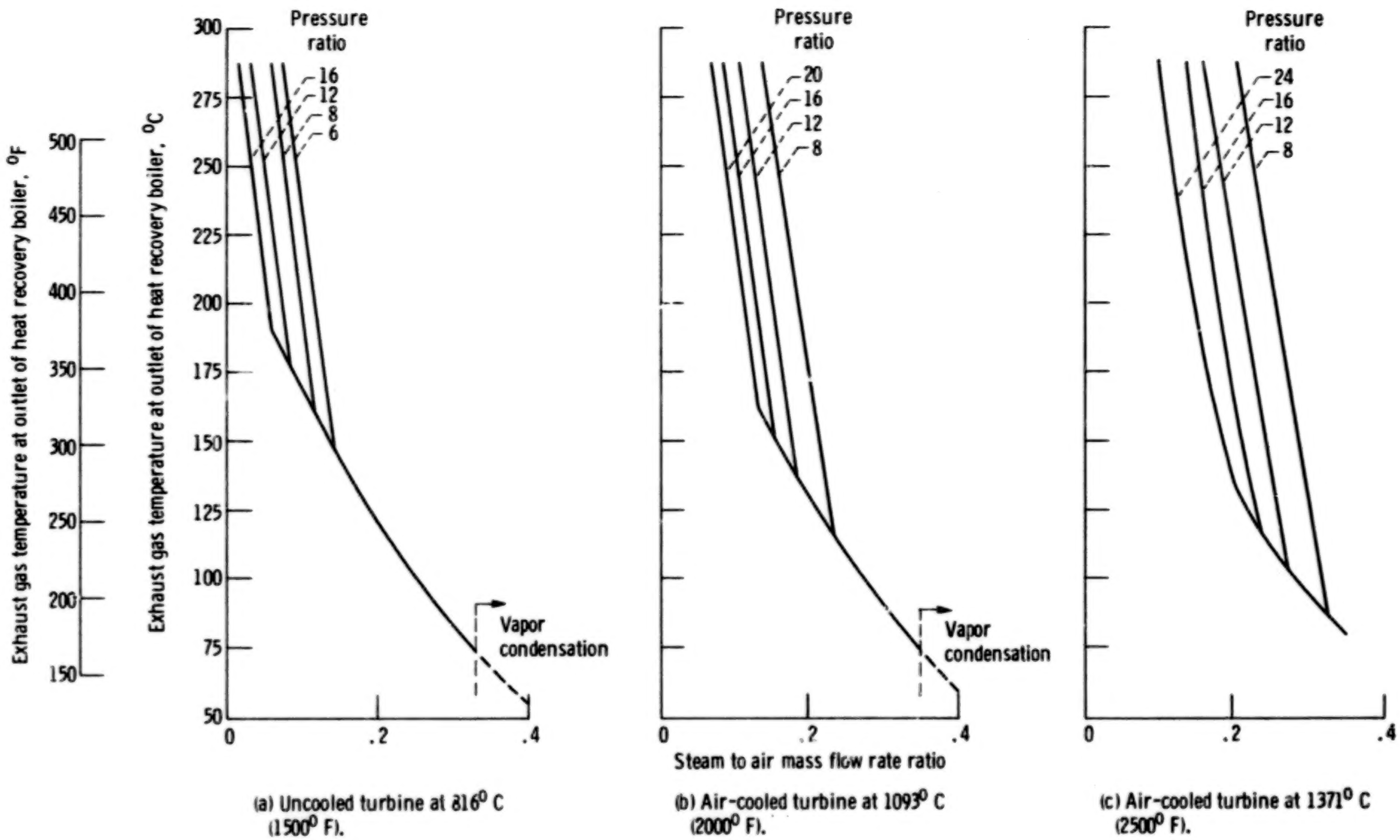
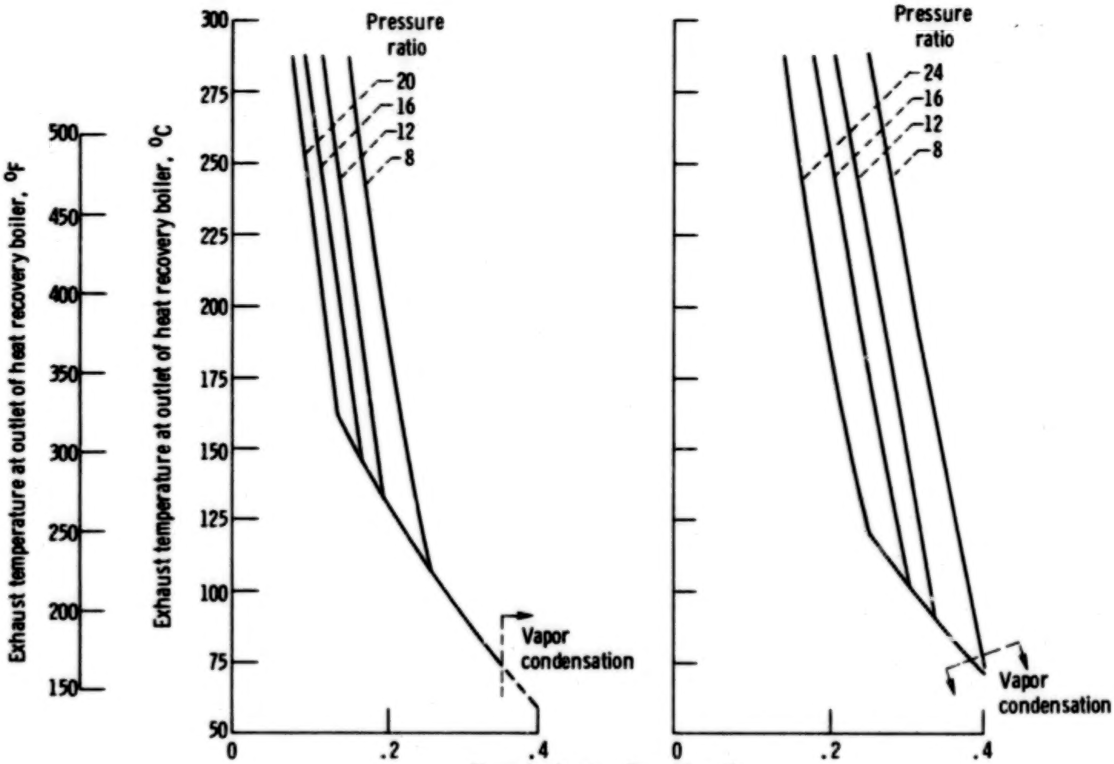


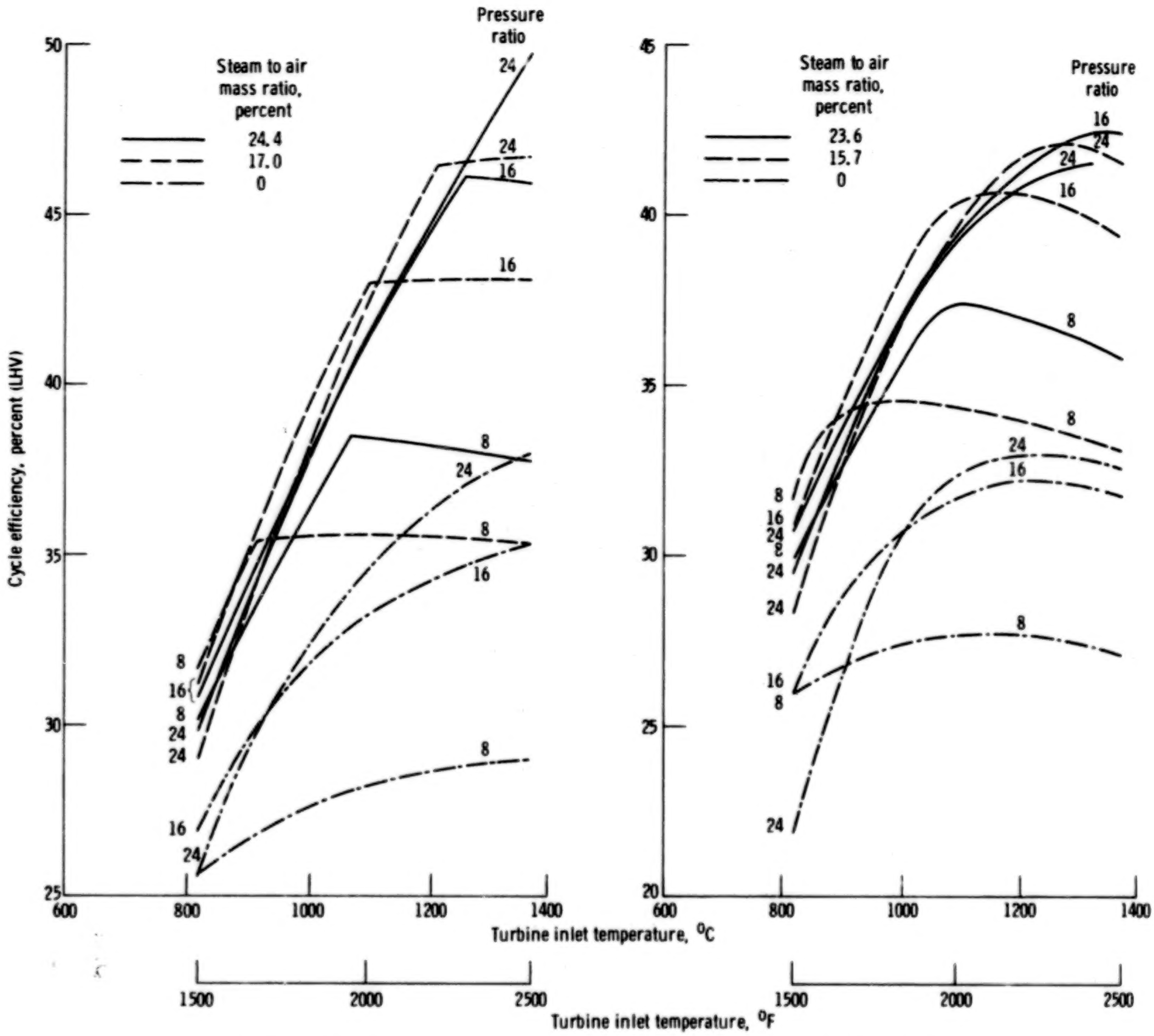
Figure 6. - Temperature of exhaust gas at outlet of heat recovery boiler as function of steam to air ratio.



(d) Uncooled turbine at 1093° C (2000° F).

(e) Uncooled turbine at 1371° C (2500° F).

Figure 6. - Concluded.



(a) Uncooled turbine.

(b) Air-cooled turbine.

Figure 7. - Effect of turbine inlet temperature on cycle efficiency for simple cycle with steam addition.

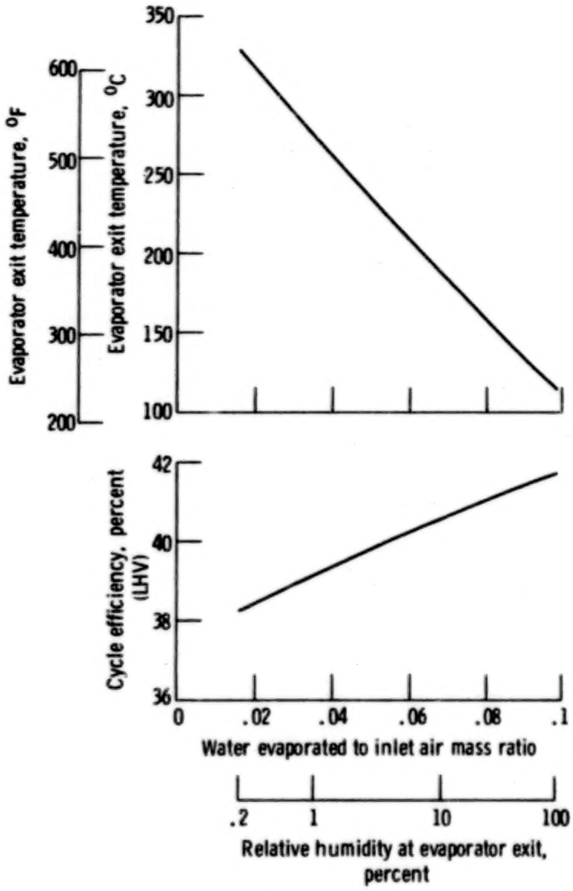


Figure 8. - Cycle efficiency and evaporator exit temperature as function of amount of water evaporated. Turbine inlet temperature, 1093° C (2000° F); uncooled turbine; pressure ratio, 12; recuperator effectiveness, 0.85.

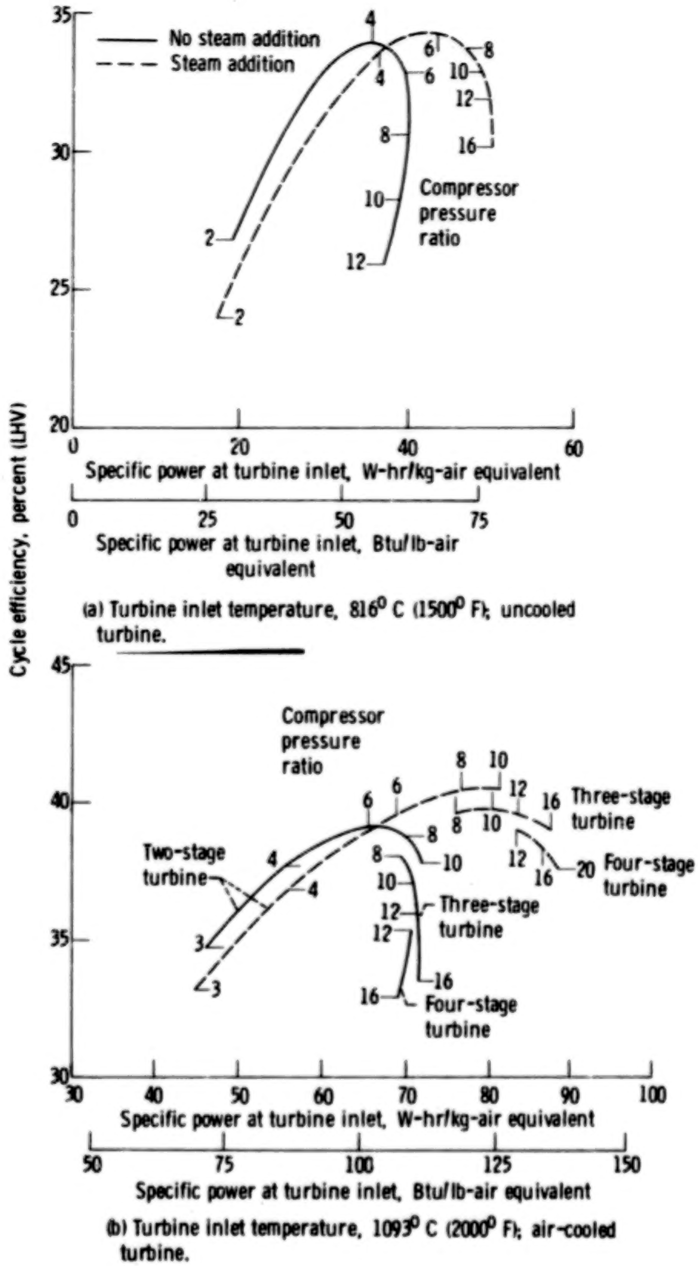
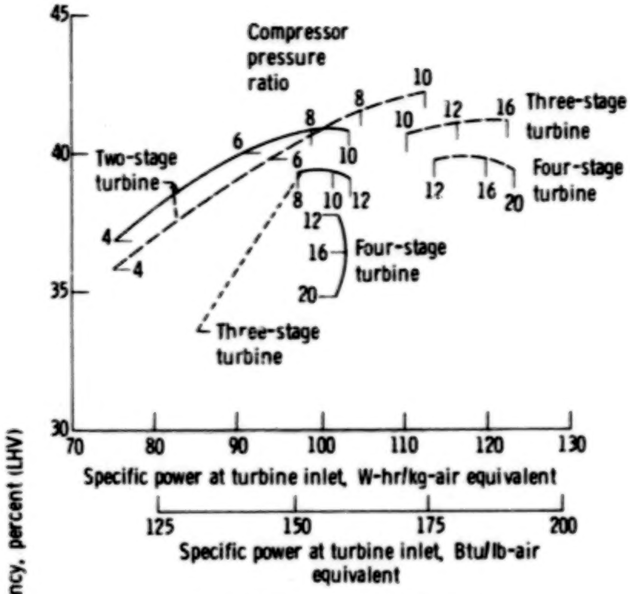
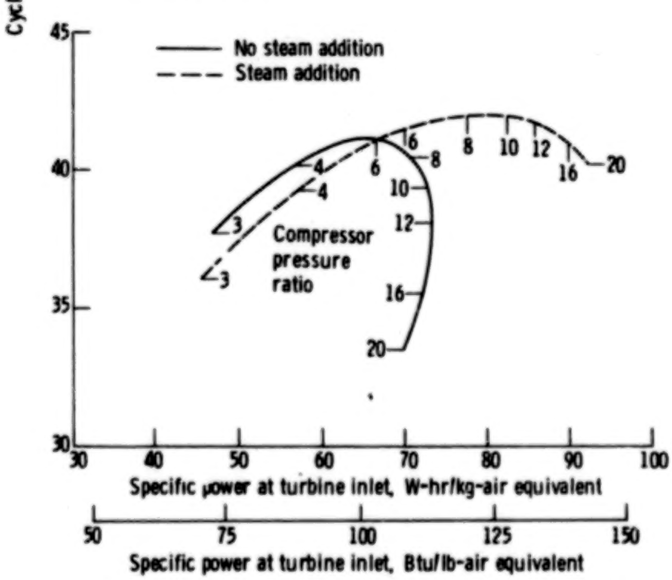


Figure 9. - Cycle efficiency and specific power for recuperated cycle with steam addition; recuperator effectiveness, 0.85.

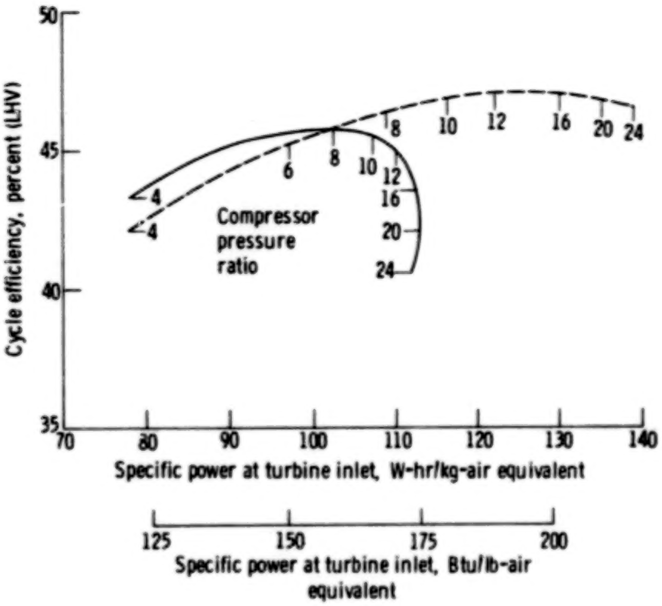


(c) Turbine inlet temperature, 1371° C (2500° F); air-cooled turbine.



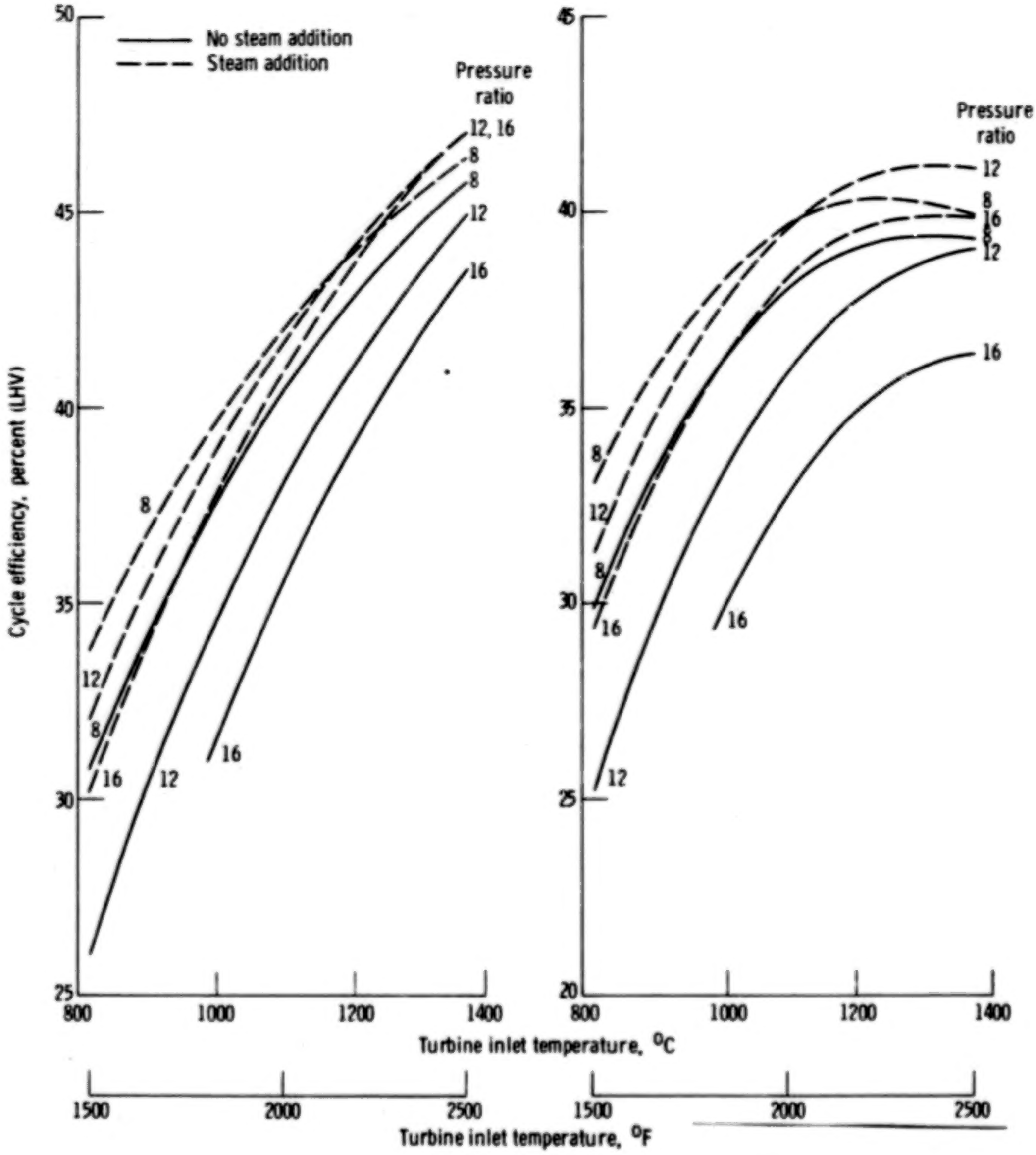
(d) Turbine inlet temperature, 1093° C (2000° F); uncooled turbine.

Figure 9. - Continued.



(e) Turbine inlet temperature, 1371° C (2500° F); uncooled turbine.

Figure 9. - Concluded.



(a) Uncooled turbine.

(b) Air-cooled turbine.

Figure 10. - Effect of turbine inlet temperature on cycle efficiency for recuperated cycle with steam addition. Recuperator effectiveness, 0.85.

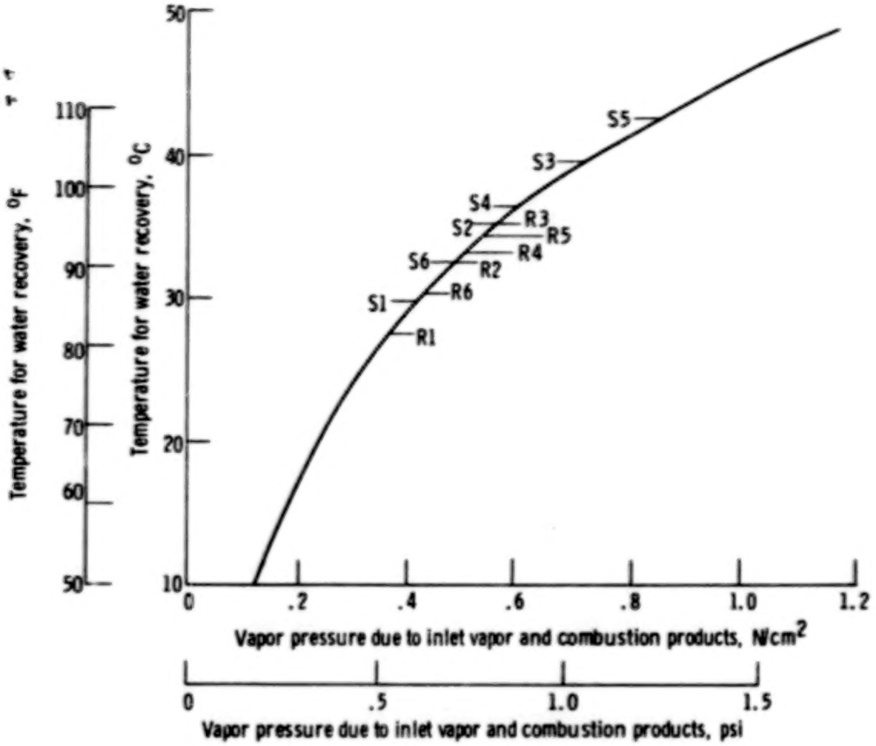


Figure 11. - Temperature and partial pressure at which all steam generated can be recovered for different simple and recuperated cycle cases.

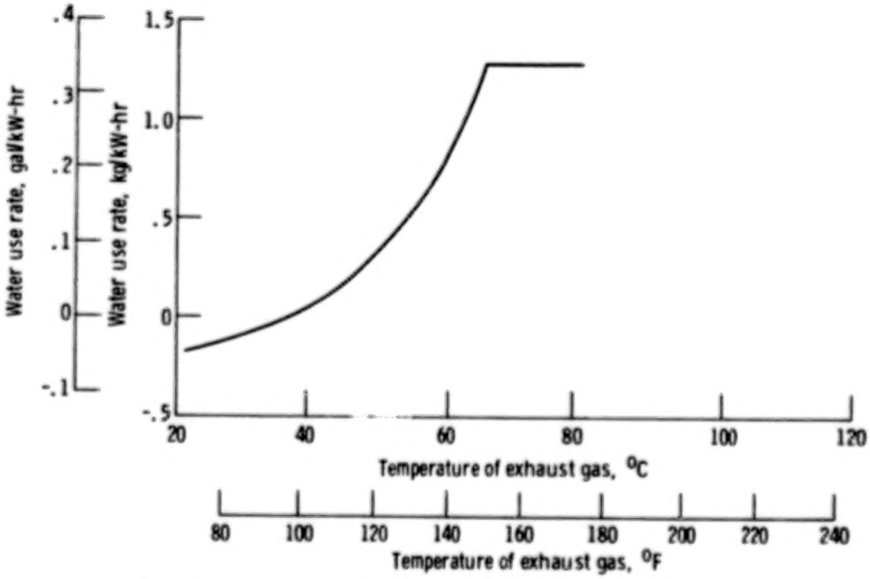


Figure 12. - Water use rate as function of condensing temperature for simple cycle with steam addition case S4.

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15. Supplementary Notes

16. Abstract

Results are presented for the cycle efficiency and specific power of simple and recuperated gas turbine cycles in which steam is generated and used to increase turbine flow. Calculations showed significant improvements in cycle efficiency and specific power by adding steam. The calculations were made using component efficiencies and loss assumptions typical of stationary powerplants. These results are presented for a range of operating temperatures and pressures. Relative heat exchanger size and the water use rate are also examined.

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